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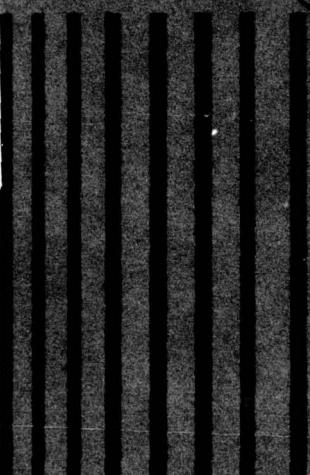
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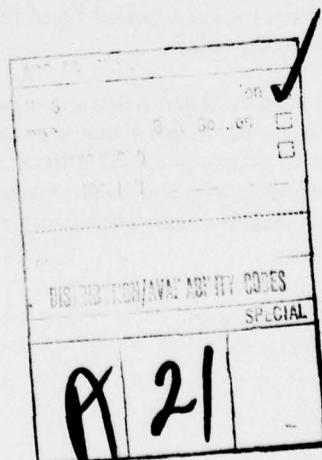
DIRECTOR NOTES

Each year I am impressed by the interest and enthusiasm of those attending the shock and vibration symposia. This was particularly true of the 48th Symposium held at the Von Braun Civic Center in Huntsville, Alabama this past October. The convention facilities were excellent. The technical program was well received. Most important, the favorable feedback from the participants on the usefulness of the Symposium has been very gratifying. Our host, the U.S. Army Missile Research and Development Command, provided outstanding support resulting in a highly successful meeting. Mr. James Daniel, MIRADCOM member of the Technical Advisory Group to SVIC, was responsible for the support requirements. He deserves high commendation and our deepest thanks.

Mr. Daniel was also Chairman of an exceptional opening session. Following a gracious welcome by Major General Charles F. Means, Commander of MIRADCOM, Dr. John L. McDaniel gave an inspiring keynote address. Dr. McDaniel recently joined Hughes Aircraft Company following his retirement as Deputy/Technical Director of MIRADCOM. The three invited speakers all gave outstanding presentations. Thanks are due to Colonel John L. Cannon, Commander of the U.S. Army Waterways Experiment Station; to Mr. E.J. Kolb, Principal Technical Information Officer for the Army from the U.S. Army Materiel Development and Readiness Command; and to Dr. Robert M. Hamilton of the U.S. Geological Survey.

With the passing of the 48th Symposium and with this issue of the DIGEST, another year is completed. SVIC looks forward to continuing service to the technical community. For now, I extend my sincere best wishes to all our readers for a happy and prosperous holiday season.

H.C.P.



EDITORS RATTLE SPACE

DECLINING ATTENDANCE AT TECHNICAL MEETINGS

It seems to me that attendance at technical meetings is continuing to decline. In fact, it is not uncommon for the authors in a session to talk only to each other! In some conferences the speakers and session chairmen outnumber the audience. The reasons for this decline, I believe, are overpublication, presentation of irrelevant material, reduced technical motivation, and economics.

In editorials during this past year I have stressed the problem of overpublication: much of the technical literature is no more than a rehash of previous work. Basic technology has been well established in many engineering areas; continued republication of the same material in slightly altered form does not motivate people to attend meetings. Publication of irrelevant material -- whether it is a super-technical treatise, technology with no practical application, or solutions to trivial problems -- is next to worthless.

Motivation for seeking new technology seems to be declining for two reasons. The first: engineers have discovered that they can solve given problems with the technical expertise they already have. The second reason also has to do with the engineer: a number of technically ill-equipped practicing engineers either are not aware of their problem or are not motivated to seek help -- until they have trouble.

Economics plays a big role in attendance at meetings. In a growing economy employers are more willing to spend money on "frills" such as technical meetings. When new technology is required to develop a product, employers are willing to support the learning process. However, in the absence of new development, they are reluctant to look at the long term education of an employee. It is unfortunate when an engineer has to perform at an optimum level on short notice -- the costs involved more often than not exceed those that would have been expended in a long-term educational program.

In order to stop the decline in meeting attendance, I believe we are going to have to select more carefully the material that is presented. This can be accomplished in part by establishing guidelines for the material to be presented at meetings and by upholding those guidelines in the review process. In addition, the effort to educate employers and employees (engineers) about the value of long-term learning should be intensified!

R.L.E.

SHIPBOARD SHOCK ENVIRONMENT AND ITS MEASUREMENT

M. W. Oleson and R. O. Belsheim*

Abstract - This paper contains a review and description of ship shock environments caused by adjacent explosions. The responses of a ship's structure and equipment to these environments are also discussed.

The ability to develop a wholly satisfactory characterization of the mechanical shock environment produced by a non-contact underwater explosion in proximity to a surface ship is limited. The shock environment of equipment is influenced by several factors. In addition to the obvious effects of charge size and distance of the explosion from the ship (attack geometry), the other effects are the response of the ship's structure to underwater shock and the dynamic properties of the equipment and the ship's structure. Reasonable experimental procedures for characterizing the free-field shock wave [1] and resulting motions of the ship's structure exist. A completely satisfactory characterization of the dynamic properties of the ship's structure has not yet been formulated.

SHOCK ENVIRONMENT

About 50 percent of the energy in an underwater explosion is propagated outward from the point of detonation in the form of an underwater shock wave. To an observer at some stationary point in the water, this wave, traveling at almost 5,000 feet per second, would appear as a pressure transient with an exponential waveshape and would be of very short duration.

The remaining energy released by the explosion is contained in a highly compressed gas bubble at the point of detonation. The bubble expands and contracts in an oscillatory fashion as it floats upward and ultimately vents at the water surface. Two effects are associated with bubble pulsation: first, water in the vicinity of the gas bubble undergoes oscillatory motions as a result of volume displacement; second, shock waves of successively lower energy are generated as the bubble contracts. Although these later effects may be important in overall ship strength computations, they are not usually

significant factors of inboard shock environment. Inboard shock environment is affected by the size and position of the explosive with respect to the ship, however.

Conditions that would result in lethal hull damage are beyond the scope of this article, which concentrates on the effects of small conventional explosives at close range, large nuclear charges at long range, and various combinations of the two (Fig. 1).

Superficial equivalence between attack geometries might be based on pressure-time impulse at the target. The free-field impulse varies inversely with the distance of the explosive from the ship. The effective impulse at points near the water surface is also influenced by a surface-reflected rarefaction wave, which, in combination with the direct pressure wave, abruptly reduces the net pressure to zero. For large charges, the pressure decay is comparatively slow, and at shallow attack angles the surface cut-off effects a reduction in the free-field impulse.

Loading on a target ship varies as a function of the attack geometry. With small charges close to the ship, target loading tends to be localized -- decreasing in severity at points on the hull away from the point closest to the charge. With very large charges much farther from the ship, the shock wave is more nearly planar, and all points on the hull are loaded almost equally.

The energy in the shock wave that is transferred to the ship's hull is initially manifest as kinetic energy of motion. As the ship begins to move, restraining forces come into play. In the horizontal direction, motion is restrained by the inertia of the water on the far side of the ship. In addition, an impulsive load of opposite phase occurs when the pressure wave has propagated to the far side. In the vertical direction, motion is restrained by gravity plus unbalanced air pressure due to cavitation beneath the ship's bottom as it moves upward in response to the initial velocity. Response in the vertical direction is

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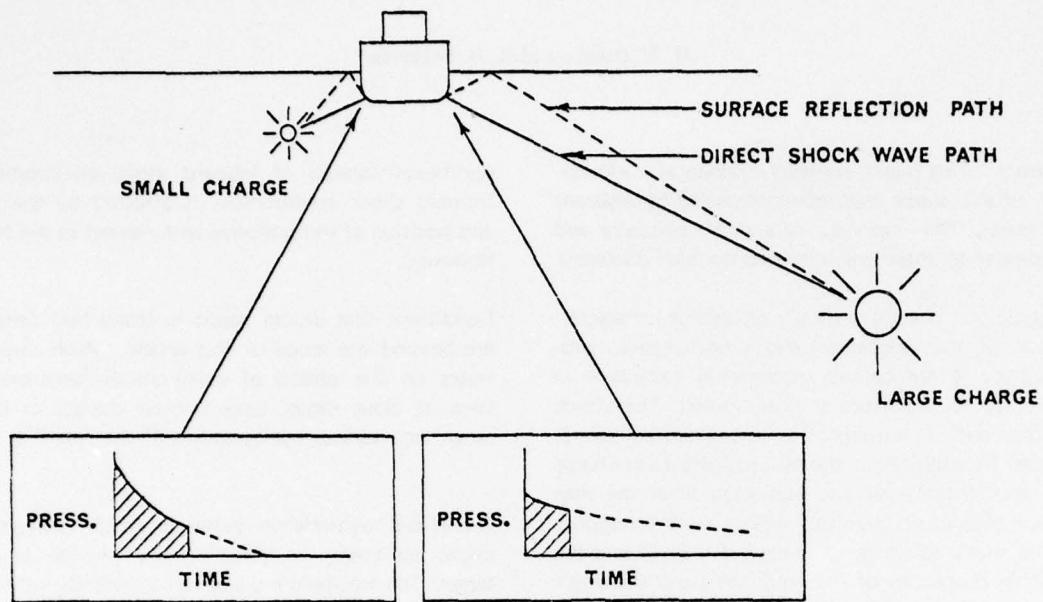


Figure 1. Pressure Time Impulses

usually greater, by a factor of two to four, than that in the horizontal direction.

SHIP RESPONSE

If a surface ship were truly rigid -- that is, without structural flexibility or structural modes -- it would respond to underwater shock as shown in Figure 2. An impulsive load from the shock wave would produce an initial-peak sawtooth velocity waveshape and a near-parabolic displacement waveshape.

Of course, surface ship's structures do have structural modes. The mode frequencies for a typical large ship range upward from one Hz, which is the first beam whipping mode [2]. Part of the kinetic energy initially transferred to the ship's bottom is manifest as rigid body motion; the remainder cause oscillatory distortions of the ship's structure at the various structural modes.

A two-mode representation of the midship's cross section amidships of a surface ship is shown in

Figure 3. The response of each mass to an impulsive load applied to the lower mass (M1) could be represented by superimposing an oscillatory component on a sawtooth velocity waveshape similar to that of the rigid mass. In other words, some portion of the incident energy has been coupled to a non-rigid mode.

The response motions indicated for this simple model are not inconsistent with experimental measurements taken during shock tests. The waveshape of the lower mass (M1) is characteristic of waveshapes taken in the hold region of surface ships. The oscillatory motion of the upper mass (M2) is frequently seen at upper deck levels. Oscillatory motion in the hold region tends to be less obvious than that indicated in Figure 3, but spectral decomposition of actual records indicates that it is present in most cases.

This two-mass representation is of course a very simplified version of a ship's structure. Mass and elasticity in a ship are distributed in structural frame members, structural hull and deck plating, and attached machinery. Although the resulting modes

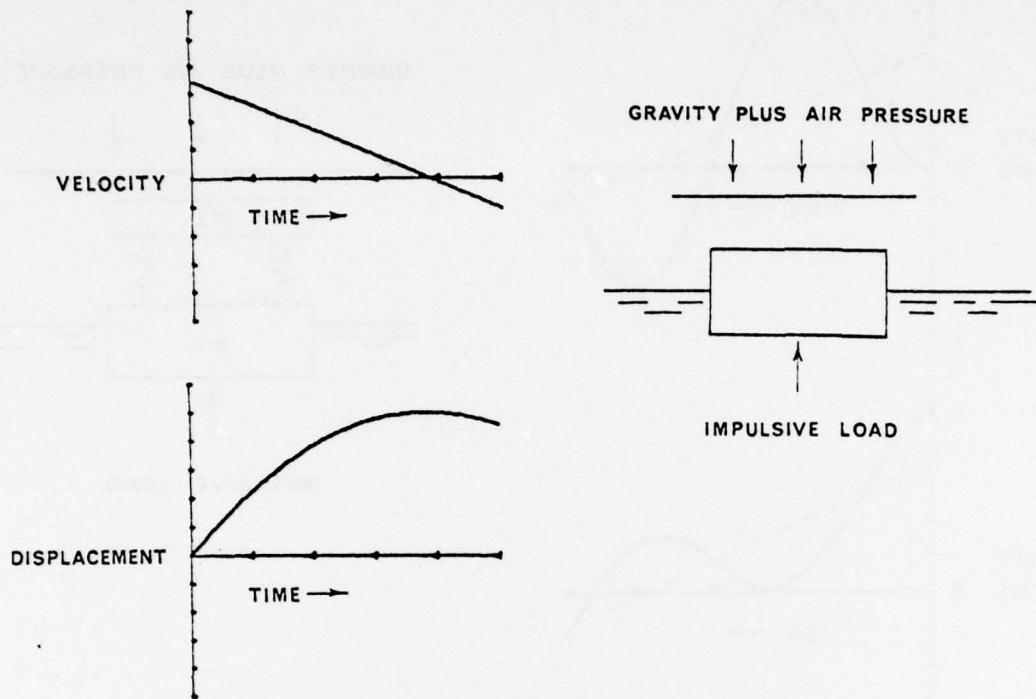


Figure 2. Response of a Rigid Structure to Underwater Shock

and their corresponding mode shapes are difficult to define both analytically and experimentally, experimental data tend to show at least one dominant lower frequency mode amidships in most surface vessels. It generally is in the range of 10 to 30 Hz and tends to have a nodal line spacing comparable to the beam of the ship.

Figure 4 is a structural schematic diagram of the cross section of amidships of an 18,000 ton combat support ship. Shock tests were conducted by placing a large conventional charge off the starboard beam. The response of the ship's structure was measured at port, centerline, and starboard positions below the main deck and at centerline positions above the main deck.

The velocity waveshapes shown in Figure 5 are positioned in approximately the same physical way as the gages in the ship. They show the first 100 milliseconds of response motion. Note the comparatively steep leading edges of the velocity waveshapes in the hold region. Note also that the initial steep rise is successively delayed at the centerline and port positions. The delay times correspond to the propagation time of the shock wave as it passed below the ship's hull. With respect to the transit time of the shock wave, therefore, the hold region was dynamically flexible.

As shock energy was propagated upward in the ship's structure, higher frequency motion components were attenuated by structural modes of the ship, and

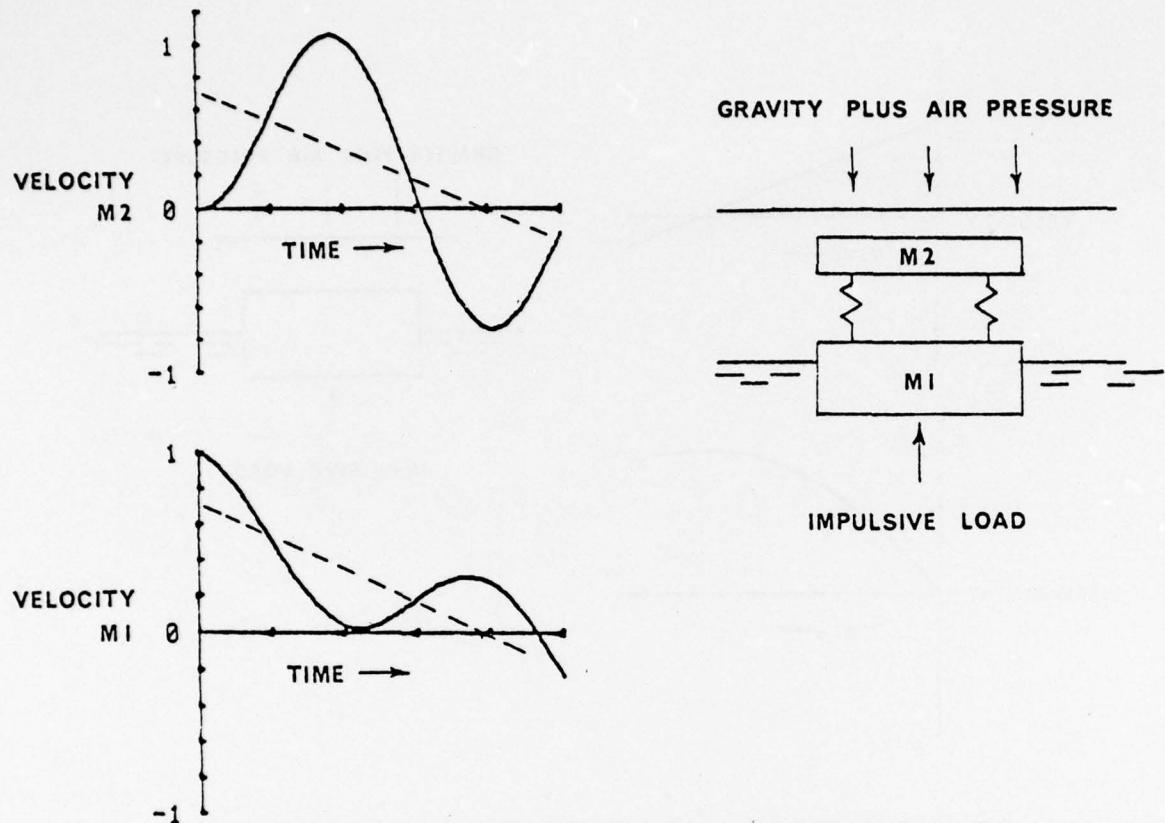


Figure 3. Two-Mode Representation of the Cross Section Amidships of a Surface Ship

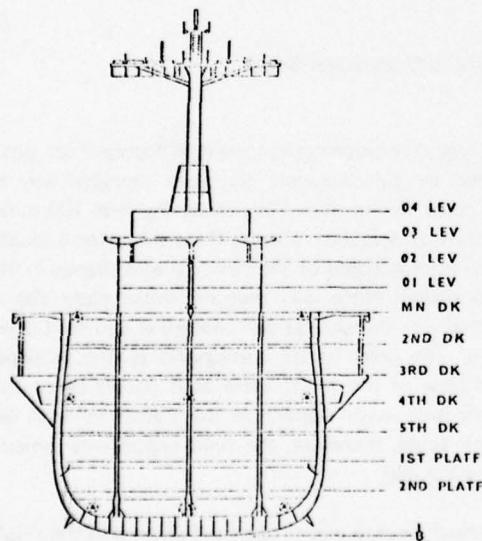


Figure 4. Structural Schematic Diagram of a Cross Section Amidships of an 18,000 Ton Combat Support Ship

responses at lower frequency modes became more prominent. As a matter of fact, if only the centerline gages are considered, it would not be difficult to justify a simple two-mode representation of the structural response of the ship. The upward velocity maximum measured at the 02 level and at the mast positions is approximately out of phase with a less obvious oscillatory component measured in the hold. The inadequacy of a two-mode representation would become evident if it were used to account for velocity waveshapes at port and starboard gage positions.

High frequency components of structural motion are most evident in the hold, as might be expected. Physically, this region of the ship is most affected by the incident shock wave. The shock wave loading is potentially capable of driving structural modes from the lowest value to the highest. However, the higher frequency modes tend to have comparatively closely

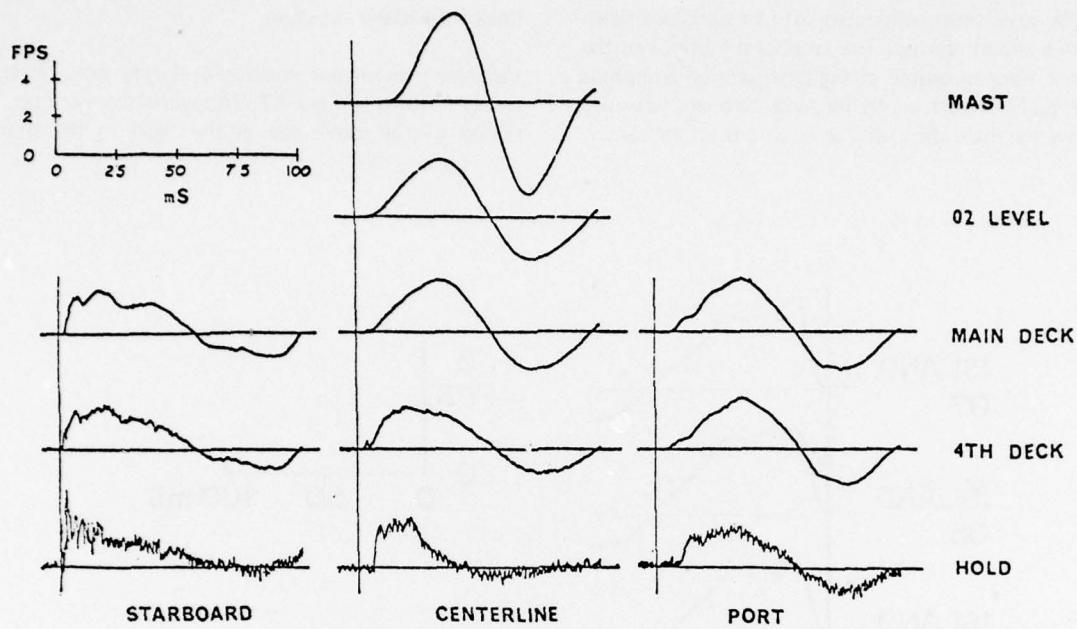


Figure 5. Velocity Waveshapes of a Ship's Response to Shock Tests

spaced nodal lines -- that is, they involve small regions of the ship -- and energy in these modes is not readily propagated over large regions of the ship. Thus, at instrumented positions on upper deck levels, the higher frequency modes were less vigorously excited, and higher frequency motion components appear progressively attenuated.

In a sense, a ship's structure can be viewed as a mechanical low-pass filter. Much equipment would experience a less severe shock environment at the upper deck levels.

Figure 6 is a structural schematic diagram of the cross section amidships of a 28,000 ton aircraft carrier of World War II vintage. It is only partially representative of modern carrier design. Typically, carrier design differs from that of smaller ships in several ways: the superstructure is displaced to one side; interior framing is interrupted at the main, or hanger deck, level; the cross section amidships is more nearly rectangular; and the multiple side tanks tend to increase the vertical stiffness of the port and starboard sides of the hull.

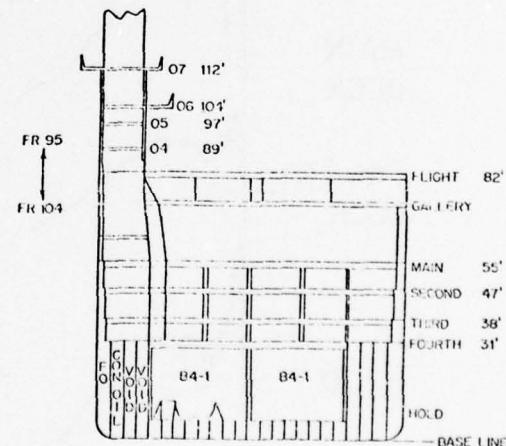


Figure 6. Structural Schematic Diagram of the Cross Section Amidships of a 28,000 Ton Aircraft Tanker

Shock tests were conducted off the starboard beam of this aircraft carrier. The structural motions of the carrier were measured in the cross section amidships with gages at port, centerline, and starboard positions below the main deck, and at positions up the center-

line of the island structure.

Velocity waveshapes measured during one carrier test are shown in Figure 7. The waveshapes are positioned in the same way as the gages in the ship.

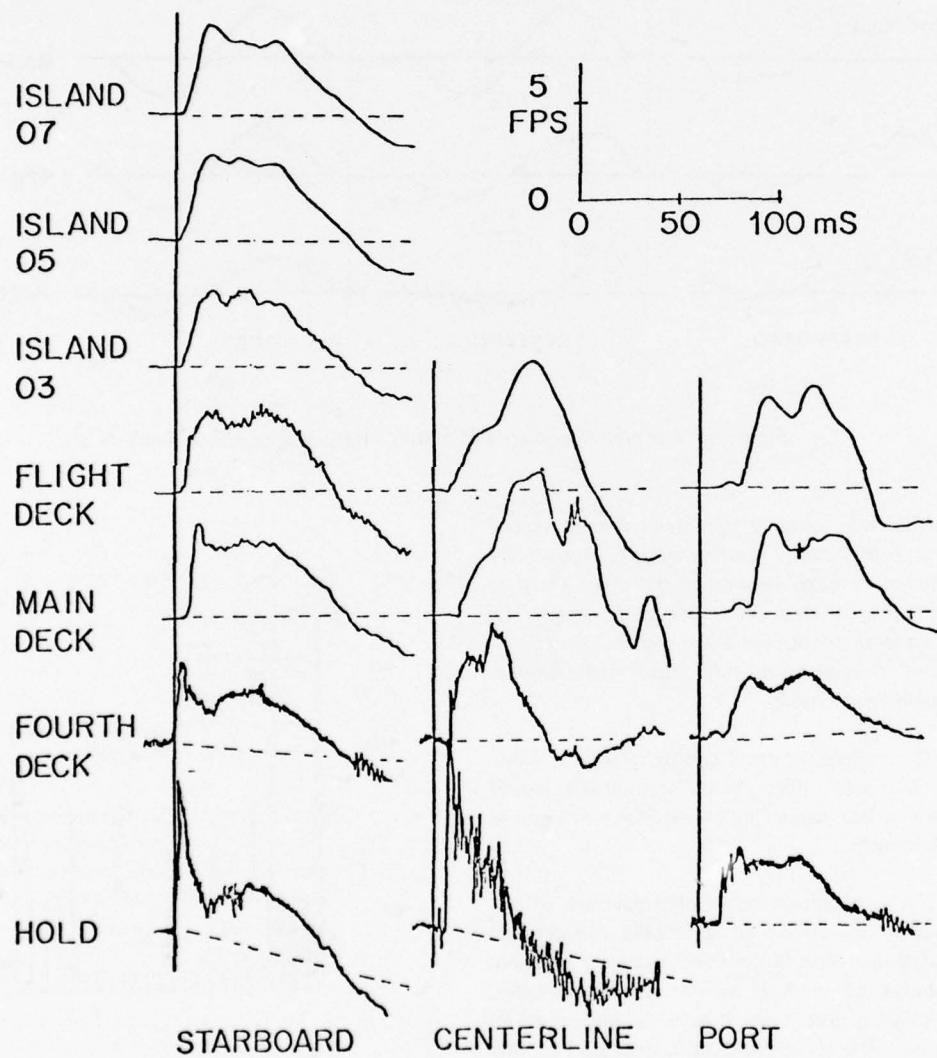


Figure 7. Velocity Responses of a Carrier to Shock Loading

The dominant features of these waveshapes are not unlike those of the smaller ship. Velocity waveshapes in the hold region have a steep leading edge and more high frequency motion than is evident at upper levels in the ship. At the centerline gage positions, motion at upper deck levels is almost sinusoidal, and would appear to be nearly out of phase with an oscillatory component of motion at the centerline gage position in the hold. This set of waveshapes has allowed an approximate experimental definition of a major structural mode of the ship.

At the flight deck level, the velocity waveshape at the centerline position is almost sinusoidal; however, the velocity waveshapes at the port and starboard edge positions might better be approximated as rectangles. The velocity waveshapes at positions on the main deck level are similar. The difference in waveshapes implies relative deflection of the centerline gage positions with respect to the port and starboard edge gage positions.

Recordings from each of three sets of gages were electronically combined and integrated to provide a time-history record of the relative deflection at deck centerline positions with respect to a line drawn between the two deck edge gage positions [3]. The time-history deflection records indicated sustained oscillatory deflections with frequencies in the neighborhood of 10 to 15 Hz.

This analysis, and other supporting data lead to the conclusion that the structural response mode involves vertical oscillation of the centerline region with respect to the sides. Such a mode might involve a significant fraction of the total mass of the ship. Because the mode was strongly excited by the incident shock, it would also contain a significant portion of the incident shock energy.

Another consequence concerns the shock environment of shipboard equipment. In most cases, damage to equipment can be related to shock-induced distortions within the equipment at natural frequencies of the equipment itself [4]. Energy to produce such distortions must necessarily be introduced via the ship structure. If a structural mode and a natural frequency of a piece of equipment were approximately the same, and if the structural mode contained significant energy, it would seem probable that the equipment damage would be enhanced.

A distinguishing feature among combatant ships of various classes regarding the shock environment of inboard equipment is associated with structural modes. In principle structural modes can be calculated. In practice such calculation has not proven adequate, and the characterization of inboard shock environment for various classes of combatant ship has been based on experimental data taken during ship-shock tests.

A broader concern is not with structural response motions of the ship alone but rather relates to the potential for damage to vital equipment.

A technological objective is to quantify and define, to engineering accuracy, shock-induced mechanical stress effects on arbitrary shipboard equipment. The engineering calculation as it pertains to the equipment is not especially difficult -- provided only an appropriate input motion (or design equivalent) can be stipulated at the equipment's foundation.

It is tempting to assume that an input motion could easily be synthesized by using a motion characteristic of the structural response of the ship. Unfortunately, such a formulation has limited validity.

The susceptibility of shipboard equipment to damage from shock is a function of natural frequencies of the equipment and of the ship's structure as well as the severity of the shock. If frequencies of the equipment are relatively high compared to the frequencies of contiguous motions of the ship's structure, stresses within the equipment can sometimes be approximated by using the weight of the equipment and measuring a peak acceleration value at the equipment foundation. Conversely, if equipment frequencies are relatively low -- a situation encountered with shock-mounted equipment -- stresses within the equipment can sometimes be approximated on the basis of excursions appropriate to the equipment foundation. For equipment whose structural frequencies are in the same general range as those of the structural modes of the ship, no single or simple parameter suitably characterizes the effective severity of the shock environment on the equipment.

In general, shock-induced response motions of installed equipment cause corresponding reaction forces at the equipment foundation. These reaction forces, in turn, tend to modify the input motion at the

equipment foundation over that which would be observed were the equipment not in place -- usually in such a way as to reduce the response motions and corresponding stresses in the equipment [5].

In Figure 8, a measured shock velocity response has been transformed from the time domain to the frequency domain with a shock spectrum analysis. The ordinate is a measure of the response that a simple mechanical oscillator would exhibit at each frequency along the abscissa axis.

The velocity record from which this figure was derived was measured at the foundation of an 8,000 lb. mechanical mass-spring assembly attached to

heavy deck plating on the third deck of a cruiser. The vertical arrow indicates the fundamental frequency of the mass-spring system, 47 Hz.

It can be shown that the maximum stress in the mass-spring system was directly related to the shock spectrum value measured at its natural frequency [6]. Yet the shock spectrum value at this frequency is greatly depressed with respect to the value at other frequencies. In effect, the assembly has reacted on foundation structure of the ship in such a way as to lessen the effect of the shock. The mass-spring system is artificial, however, because it was designed and installed to demonstrate the effect of structural interaction.

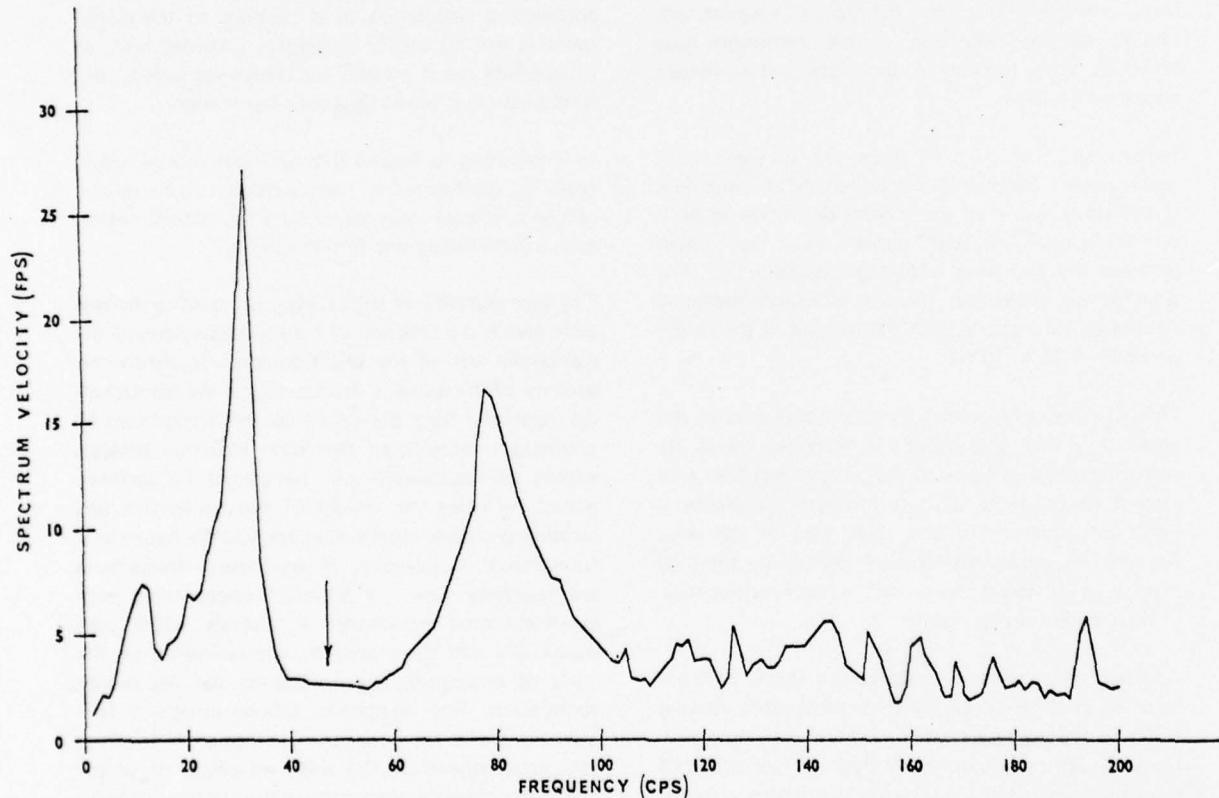


Figure 8. Shock Velocity Response Spectrum

The spectrum in Figure 9 was obtained from a velocity response measured at the foundation of an actual service turbo generator of a ship (SSTG). The SSTG weighed 33,000 pounds and was attached to the cruiser hull by vertical pipe stanchions. The SSTG installation exhibited a single dominant natural frequency at almost 30 Hz. Again there is clear evidence of structural reaction on the ship -- an obvious depression of the shock spectrum occurring at the natural frequency of the installed equipment.

In general, the effect of structural interaction is determined by modal frequencies and modal weights of both the equipment and the adjacent ship structure. Lightweight equipment attached to a heavy ship structure would probably cause little modification of the structural response of the unloaded ship.

Conversely, comparatively heavy equipment could be expected to produce a greater mitigation influence on its own environment.

The need to account for such structural interaction poses a substantial complication in the effort to characterize shock environments of ships. Structural dynamics as well as structural motion of the ship must be characterized.

A similar complication in electrical network analysis is readily reduced by means of Thevenin's theorem -- one of several network theorems that apply to linear systems. On the basis of this theorem, a complicated electrical network of active sources and passive circuit components can frequently be represented by a single equivalent source and a single equivalent circuit impedance. A similar representation of a ship structure might involve an equivalent velocity-time history and an equivalent mechanical mobility at selected positions on the ship's structure. Proposals to develop such a characterization have been made in past years, but have not been implemented [7]. Indeed it is not clear that current technology is adequate to accomplish the task.

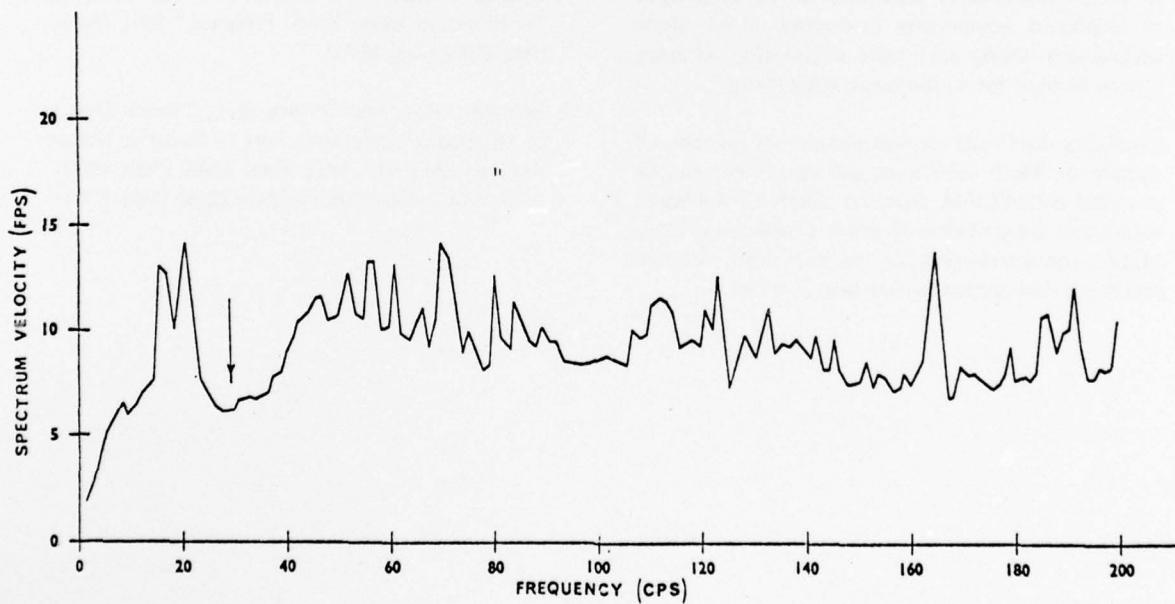


Figure 9. Velocity Response Spectrum at the Foundation of a Ship's Service Turbo Generator

Another approach to adequate characterization of the ship's structure is implicit in the Navy's Dynamic Design Analysis Method (DDAM). Numerical values used for DDAM calculations have been derived from motion measurements taken at the foundations of installed equipment [8]. Such measurements account for both structural interaction and basic structural response of the ship. Properly interpreted, the motion measurements are appropriate to other, similar installations.

DDAM in its present form, however, is not adequate for an engineering analysis of all classes of shipboard equipment. Any experimental measurement essentially characterizes the combined influences of structural response and structural dynamics of the ship and structural dynamics of the equipment. A large number of possible combinations exists, of course. The synthesis of a large number of experimental measurements has yielded a generalized design input for many of the more important combinations. But, for practical purposes, the existing data base is not extensive enough to characterize all combinations of engineering interest.

In fact, contemporary capability for shock analysis of shipboard equipments is limited; shock stress calculations having acceptable engineering accuracy cannot be done for all shipboard equipment.

Capability does exist for evaluating shock hardness of equipment. Much vital shipboard equipment can be analyzed with DDAM. Selected classes of shipboard equipment are amenable to analysis based on existing motion characterizations of the ship shock environment, and new approaches are being studied.

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LITERATURE REVIEW

survey and analysis
of the Shock and
Vibration literature

The monthly Literature Review, a subjective critique and summary of the literature, consists of two to four review articles each month, 3,000 to 4,000 words in length. The purpose of this section is to present a "digest" of literature over a period of three years. Planned by the Technical Editor, this section provides the DIGEST reader with up-to-date insights into current technology in more than 150 topic areas. Review articles include technical information from articles, reports, and unpublished proceedings. Each article also contains a minor tutorial of the technical area under discussion, a survey and evaluation of the new literature, and recommendations. Review articles are written by experts in the shock and vibration field.

This issue of the DIGEST features a literature review on a new way to model mechanisms and machines by Dr. R.C. Winfrey. His article on the finite element method applied to the analysis of mechanisms and machines reflects a new way at looking at the problem.

Drs. Ross, Strickland and Sierakowski review experiments involving basic structural elements such as beams and plates subjected to blast loading. Responses and failures of these elements are described.

RESPONSE AND FAILURE OF SIMPLE STRUCTURAL ELEMENTS SUBJECTED TO BLAST LOADINGS

C.A. Ross*, W.S. Strickland**, and R.L. Sierakowski***

Abstract - This paper is a review of experiments involving basic structural elements such as beams, plates, and cylindrical shells that have been exposed to mild blast loadings. The response and subsequent failure of these structural elements are described in some detail.

The response and failure of structural elements under dynamic loadings are complicated processes that are difficult to analyze. The responses of beams and plates to blast loadings are similar; the response of cylindrical shells tends to be much less predictable and more complicated. This paper describes the effects of mild blast loadings on these simple structural elements.

BEAMS

Aluminum beams, 0.0254 m wide, 0.454 m long, and 0.16 to 0.32 cm (0.063 - 0.125 in) thick were exposed to a fuel-air-explosive (FAE) device. This fuel-air device, which was used in all of the tests consists of a plate and beam test fixture fabricated from 2.54 cm steel plate and bolted to a concrete pad; a gas bag containing the fuel-air mixture is placed in series as shown in Figure 1. Polyurethane plastic is stretched over a waterpipe frame, and the

assembly is sealed with plastic tape. A detonating charge of 100 gr of Data Sheet is placed at the end of the bag opposite the plate; the bag is filled with 0.91 kg of MAPP (methyl acetylene propadiene) gas and allowed to mix with air for ten minutes. Detonation of the Data Sheet creates a Chapman-Jouget wave as the fuel air mixture travels the length of the bag and impinges upon the test device. The device produces a wave of constant velocity and pressure; the reflected pressure on the test item can be varied, however, by changing the distance from the end of the bag to the test item (D of Fig. 1). Initial measurements were made on a thick non-deforming plate instrumented with piezoelectric transducers for recording peak pressure versus position for various distances between the bag and the test fixture. As a check, pressure was measured around the outside of the test section during both the plate and beam tests. Pressure and impulse data reported herein are based on pressure-time histories recorded on the flat non-deforming plate.

Both ends of the beams were held fixed against rotation and deflection. The load was applied normal to the 2.54-cm beam width by placing the bag in series with the test stand (see Fig. 1). Pressure was also measured on 4.90-cm thick steel beams fixed as shown in Figure 2. Deflection-time histories for

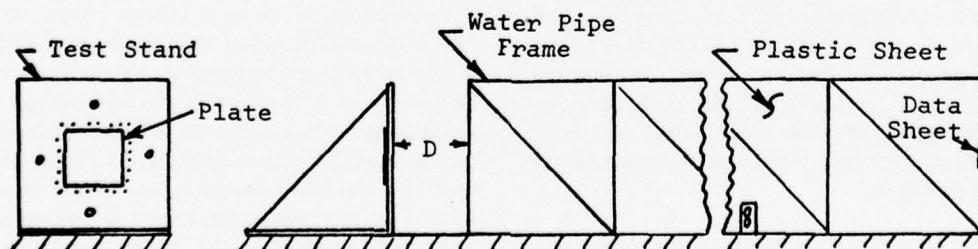


Figure 1. Gas Bag and Plate Test Fixture in Place

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***Professor, Engineering Sciences Dept., University of Florida, Gainesville, FL 32611

the beam were obtained by placing a lined grid to the side and behind the beam (see Fig. 3) and using a high-speed camera to record the deflection.

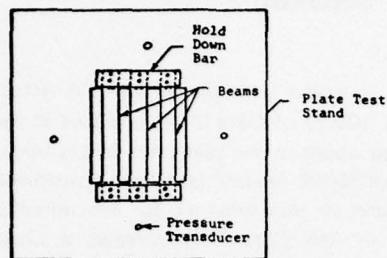


Figure 2. Beam Test Specimen Bolted in Place

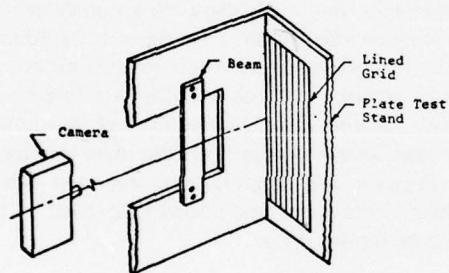


Figure 3. Test Fixture for Recording Time History of Beam Deflection

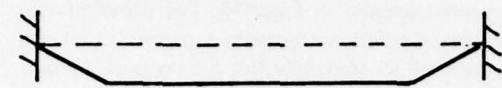
The responses of the beams can be separated according to beam thickness: with thick beams, large permanent deflection occurs with little or no rebound; with thinner beams, large deflections take place with considerable rebound. The response and failure of the thick beams (0.32 cm) can involve: (a) permanent deformation without failure; (b) failure at some critical load and deflection; (c) failure during the response mode before maximum deflection; and (d) shear failure at the edges before deformation begins.

Deflection of the 0.32-cm thick beams occurs as a traveling hinge motion (see Fig. 4a, b), which continues until the motion reaches the midpoint of the beam. If the loading is sufficient, failure can occur at the fixed ends; with smaller loads, some elastic rebound occurs, and the beam is permanently de-

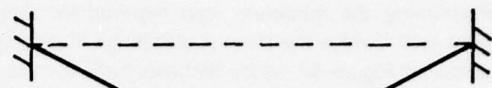
formed (see Fig. 4c). Apparently, for a given beam a critical load exists at which maximum deflection for failure occurs. For all beams tested, failure occurred at one of the fixed ends. If the load was increased beyond the critical load where failure and maximum deflection are coincident, failure occurred at the fixed ends during the initial hinge motion. Continued increases in the loading could cause shear failure at the fixed ends before any noticeable deformation took place [1]. It would appear that the failure mode for the thicker beams changes from a tensile failure to a shear failure with increases in loading at constant thickness.

Based on tests on aluminum beams [2] the traveling hinge velocity -- approximately 3,000 m/sec -- indicates a shear wave. The tensile to shear failure transition can be explained with a critical shear particle velocity concept. The initial transverse velocities of the beam were calculated from known applied impulse values and compared to the critical shear particle velocity for the beam material [3]. The initial transverse velocity exceeded the critical shear particle velocity in each case of shear failure at the fixed ends.

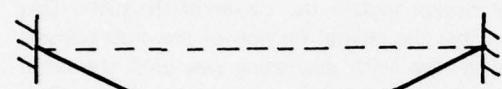
The thinner beam elements show considerable rebound if failure does not occur. Film clips obtained according to the scheme in Figure 3 showed that the beams begin to deflect with a traveling hinge motion (see Fig. 4a, b). The hinge motion continues to the midpoint of the beam (Fig. 4c) at which time a reflection of the waves occurs and the beam begins to rebound in the same shape as that of the initial deformation (note flat midsection of Fig. 4e). If the beam fails, failure occurs when the two traveling hinges reach the midpoint of the beam. If failure does not occur, rebound continues toward the initial position of the beam in a traveling hinge motion. The plastic deformation that occurred during the initial deformation increased the length of the beam, however, and it is therefore too long to pass back through its initial position without buckling. A typical buckling pattern is shown in Figure 4f. The beam thus oscillates several times through its underformed position and comes to rest in a shape similar to that of Figure 4h. As the load is increased to some critical value, failure occurs in a deflected mode similar to thick beam failure. Continued load increases beyond this critical value ultimately produce the failure described for thicker beams.



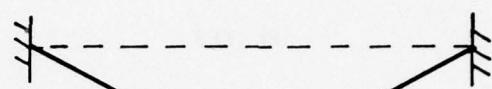
a



b



c



d



e



f



g



h

l

Figure 4. Typical Responses of Beam Elements

Based on these experimental results, a method for determining the minimum load required for failure might well involve the three plastic hinge mode shape typical of Figure 4c, which includes both the plastic bending stress and the axial stress.

The general traveling hinge motion has been described in detail for beams with and without axial restraint [4, 5]. Experimental tests [6] have shown that impulsively loaded beams without axial restraint also exhibit traveling hinge motion without rebound for all thicknesses of beams.

PLATES

Both 2024 aluminum and mild steel plates were tested by subjecting 0.46 m square plates to the FAE device shown in Figure 1. The test plates were held fixed on all edges with a friction device; post test inspection showed very little slippage at the edges.

A reflective Moire fringe pattern was used to observe the deflection-time history of selected plates (see Fig. 5). Additional information regarding this technique is available [7, 8].

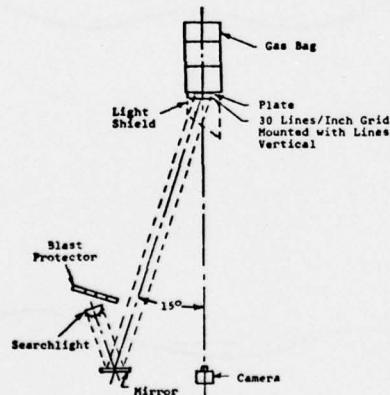


Figure 5. Schematic Diagram for Moire Pattern Experiments

Square plates 0.46 m (18 in.) wide and from 0.064 to 0.32 cm (0.025 - 0.125 in.) thick were tested at various blast pressures. The response modes for the aluminum and steel plates were similar. Reflective Moire fringe patterns photographed at 40,000 frames/

sec revealed a hinge type motion for the deflection shown schematically in Figure 6. The actual photograph cannot be shown because the contrast of the fringe pattern is completely lost during reproduction from the movie film.

A single fringe of the Moire pattern simulated in Figure 6 represents a line of constant deflection. The spacing between the fringe lines would represent the density or gradient of the deflection with respect to the normal to the fringes. As indicated in Figure 6a, the deflection starts with motion of the entire plate. The boundary is seen as a moving wave or hinge motion toward the center of the plate. This means that the central portion of the plate remains relatively flat with decreasing size until the hinge has nearly reached the center of the plate. This central flat portion retains an almost square shape through about half of the deflection process; then the central portion of the plate begins to bulge uniformly and takes on an almost spherical shape (see Fig. 6d). The center of the plate continues to deflect, and the spherical portion enlarges slightly. Failures usually occur at this point in time and begin as cracks at the midpoint along one of the plate edges. The cracks grow in both directions around the edges of the plate, cutting across the corners approximately one-quarter of the diagonal distance across the plate from the corner as shown in Figure 7. The failure surface of the crack appears to be a typical sheet failure in tension.

For the thinner plates and lower peak pressures some rebound will occur even though plastic deformation has already occurred. Although it was believed that some reverse flow from the blast was causing the rebound, high-speed photography verifies that elastic rebound does occur. This rebound was found to be more prevalent for thin beams than for plates. Table 1 lists all the plates tested, as well as pertinent data measured and recorded.

It was observed experimentally that plate failure occurs in an almost fundamental mode for the loadings used in this study, even though the higher modes are active during the major portion of the deformation process. For more severe loadings failure begins as shear of the sheet at the edges before any deformation takes place. However, this type shear failure requires that the peak pressure be greater than that for any failure occurring from

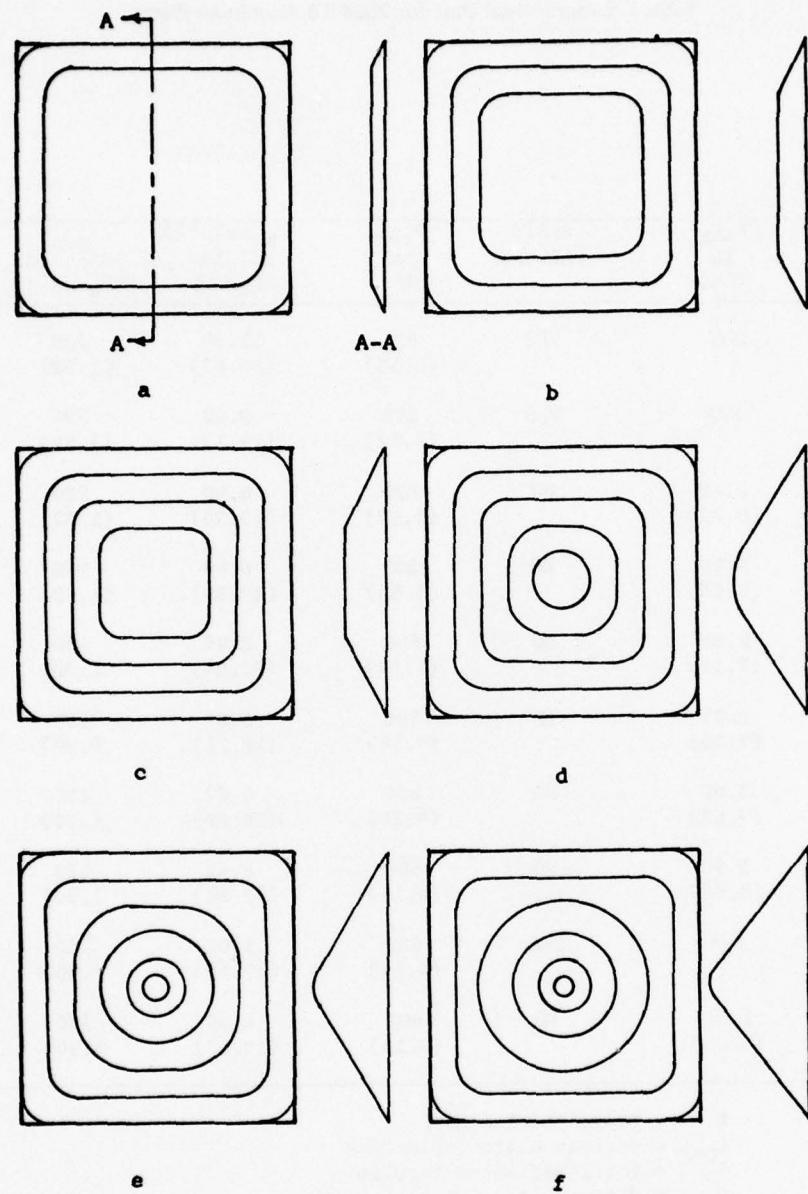


Figure 6. Sketch of Moire Fringe Patterns

The diagram to the right of each pattern represents the shape of the plate across centerline as shown at a typical section A-A of (a).

Table I. Experimental Data for 2024-T3 Aluminum Plates

h IN (CM)	w_{max} IN (CM)	PLATE FAILURE	p_{max} psi (MP_a)	(P_m/hx10⁻³) psi/in (MP_a/cm)	I_{max} psi-msec (MP_a-msec)	D FT. (M)
.071 (.180)	N/A	YES	800 (5.52)	11.30 (30.67)	220 (1.52)	0
.090 (.229)	N/A	YES	800 (5.52)	8.90 (24.17)	220 (1.52)	0
.125 (.318)	2.45 (6.22)	NO	800 (5.52)	6.40 (17.38)	220 (1.52)	0
.125 (.318)	2.60 (6.60)	NO	800 (5.52)	6.40 (17.38)	220 (1.52)	0
.071 (.180)	2.80 (7.11)	NO	600 (4.14)	8.45 (22.94)	130 (.90)	3 (.92)
.090 (.229)	2.75 (7.00)	NO	600 (4.14)	6.67 (18.11)	130 (.90)	3 (.92)
.063 (.160)	3.00 (7.62)	NO	600 (4.14)	9.52 (25.85)	130 (.90)	3 (.92)
.071 (.180)	2.70 (6.86)	NO	600 (4.14)	8.45 (22.94)	130 (.90)	3 (.92)
.050 (.127)	N/A	YES	600 (4.14)	12.00 (32.58)	130 (.90)	3 (.92)
.125 (.318)	1.98 (5.03)	NO	600 (4.14)	4.80 (13.03)	130 (.90)	3 (.92)

h = plate thickness
w_{max} = maximum plate deflection
I_m = total reflected impulse
D = distance from plate to gas bag
p_{max} = peak reflected over pressure
a = 9 in, 18 in (45.72cm) square plate for all tests
p(t) = $p_m(1-t/\tau) \exp(-\alpha t/\tau)$
 α = decay constant
 τ = positive pressure phase duration

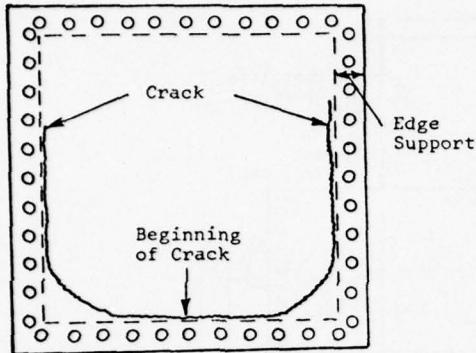


Figure 7. Typical Plate Failure

some deformation process.

The fact that, for the plates tested, failure occurs while the plate is in a fundamental mode shape supports the idea that an analysis could be based on a deformation to failure in a fundamental mode. The assumption that the energy to drive the plate to failure is independent of how it got there allows for a simple analysis. This analysis was applied to the plates tested with reasonable results for center point deflection for plates that did not fail [7]. This simple analysis also predicts failure at the midpoint of one edge when the ultimate strain, from the static stress-strain curve, is used as the failure criterion. Any strain rate effect or sensitivity is neglected, however. But, for the work hardened

material and plates tested, this assumption is not unreasonable. Figure 8 is a comparison of analytical and experimental results.

CYLINDRICAL SHELLS

Aluminum cylindrical shells with fixed ends and subjected to both a fuel air explosive (FAE) and spherical pentolite (HE) device have been studied using the test fixture shown in Figure 9. For the FAE loading the plate test fixture was replaced with the cylindrical test fixture; for the HE loading the spherical charge was hung directly over the cylinder as shown in Figure 9.

For the cylinders tested, the internal diameter was held constant at 0.31 m. Length/diameter ratios of 1.89, 0.89, and 0.39 were matched with radius/thickness ratios of 188, 117, and 95 to give a nine point data base for comparison.

The coordinate system used in the description of the response and failure is shown in Figure 10. A circumferential mode number n and a longitudinal mode number m used in the expression for radial deflection w

$$w = \sum \sum w_{mn} \cos(n\theta) \sin(m\pi x/L)$$

have been used to describe the general response modes of the cylinders. The number of buckles per circumferential length for a given mode shape

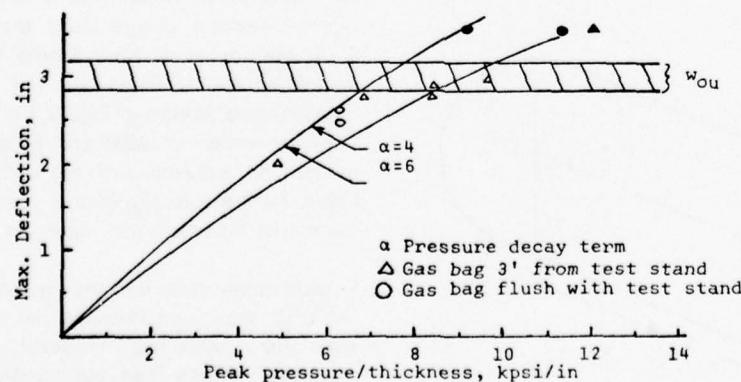


Figure 8. Maximum Plate Deflection Versus Pressure-to-Thickness Ratio

Plate failure range, w_{ou} , is based on an 18-20% ultimate strain.
Solid curves represent analytic results; solid symbols represent plate failure.

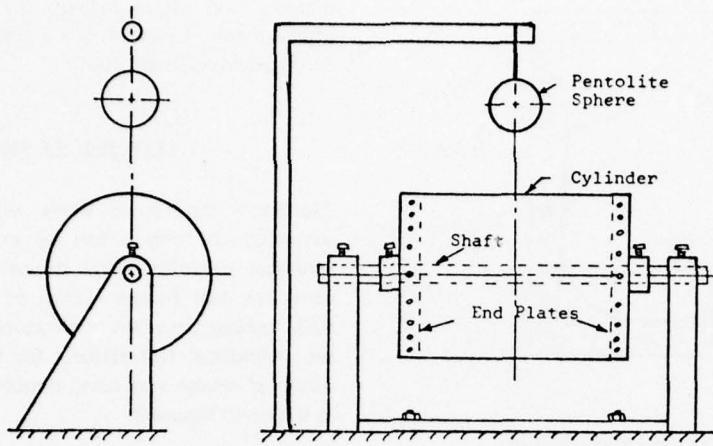


Figure 9. Cylindrical Shell Test Fixture

is n , and m is the number of half sine waves in the longitudinal direction. For all cases m was determined experimentally to be unity and is designated as the fundamental mode shape in the x direction. Experimentally n was calculated by dividing the number of buckles by the fraction of circumferential distance associated with the buckles. For example, for three buckled modes in only 25 percent of the cylinder (Fig. 11b), an experimental value of $n = 12$ is given. Attempts to photograph shell response were not successful, and all information from the FAE experiments was determined by post-test inspection. However, some high-speed photography have been obtained [9] for shock tube experiments on cylindrical shells.

With fixed end cylinders failure always began as a crack at the $\theta = 0$ position of one of the fixed ends and spread circumferentially in two directions. The failure mode of those cylinders that failed was the same regardless of the response mode shape prior to the beginning of failure.

Experiments on cylinders loaded with planar blast waves have shown that buckling begins along the length of the cylinder at the $\theta = 0$ position and spreads circumferentially to about the $\pm 45^\circ$ positions. In all cases tested the average buckled area was only about 25 percent of the circumference. Deflection of the shell coincides with buckling and forms the fundamental mode in the x direction. The maximum deflection occurs along the mid-length and $\theta = 0$ position as shown in Figure 11. In some cases circumferential buckling did not occur. The unbuckled cross section of Figure 11c is typical of this response, which is called the fundamental collapse mode. The occurrence of the various mode shapes before failure complicates the analysis and is unlike the results for beams and plates discussed previously.

Experimental data for the cylindrical shells (see Table 2) show that buckling can occur for a given shell and a given load. However, a change in the magnitude of the load can produce a collapse response (see data points 15 and 28 in Table 2). Such results suggest that, for a given cylindrical shell, some critical load provides a transition between the buckled and collapse patterns.

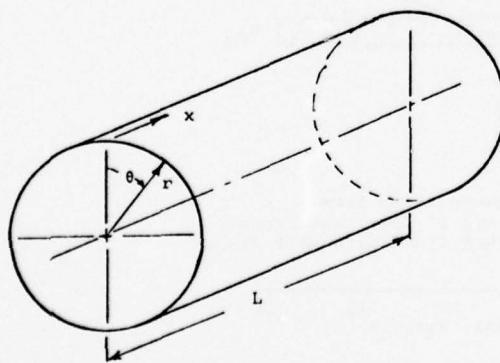
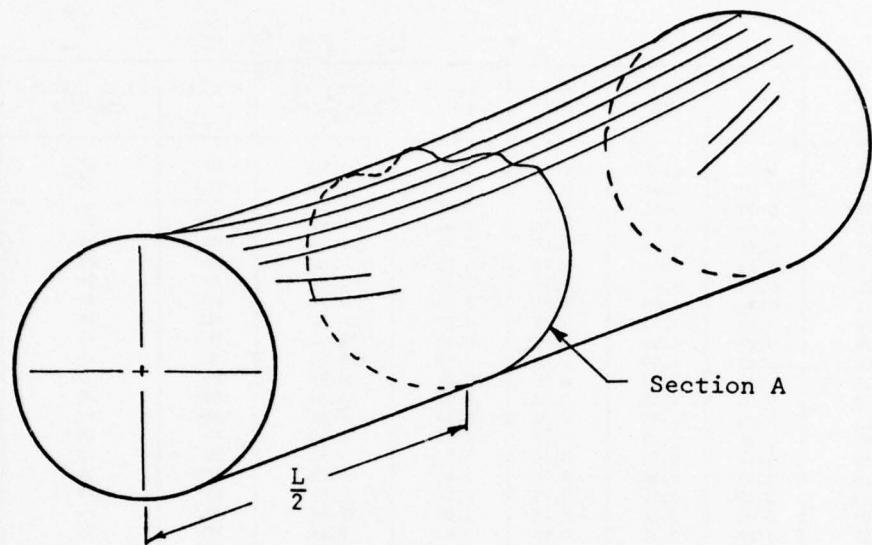
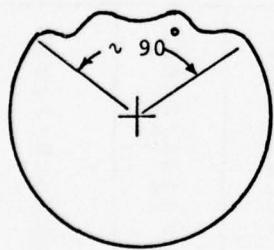


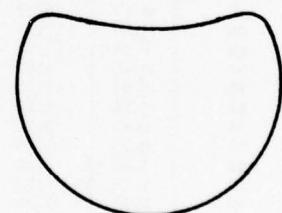
Figure 10. Coordinate System for Cylinders



a



b
Buckled Section A



c
Unbuckled Section A

Figure 11. Buckling Patterns and Modes of Cylindrical Shells

Table 2. Summary of Cylindrical Shell Tests

Fuel-Air Device

DATA POINT	a/h	L/D	P_{max} $\theta=0$	I_{max} $\theta=0$	Δt $\theta=0$	CENTER PT. DEFLECTION	FAILURE	% OF CIRCUM. DAMAGED	n
1	188	0.39	2.41	0.52	0.93	1.63	NO	30	28
2	188	0.39	2.97	0.59	0.87	2.03	YES	30	38
3	188	0.39	3.65	0.67	0.83	2.87	YES	--	34
4	188	0.39	4.48	0.76	0.80	10.41	YES	--	36
5	117	0.39	4.48	0.76	0.80	2.24	NO	32	22
6	117	0.39	6.03	0.90	0.70	2.31	NO	34	25
7	95	0.39	6.03	0.90	0.70	1.02	NO	24	26
8	95	0.39	6.03	0.90	0.70	0.74	NO	24	26
9	188	0.89	2.97	0.59	0.87	6.10	YES	34	32
10	188	0.89	3.65	0.67	0.83	>11.43	YES	--	32
11	188	0.89	4.48	0.76	0.80	>11.43	YES	--	33
12	117	0.89	4.48	0.76	0.80	3.96	NO	32	26
13	117	0.89	6.90	0.83	0.75	6.05	YES	36	34
14	117	0.89	6.03	0.90	0.70	8.26	YES	--	22
15	95	0.89	6.03	0.90	0.70	3.05	NO	34	18
16	85	0.89	6.03	0.90	0.70	2.41	NO	32	19
17	188	1.89	1.28	0.37	1.40	4.32	YES	34	13
18	188	1.89	1.52	0.40	1.20	--	NO	34	13
19	188	1.89	1.86	0.45	1.00	>11.43	YES	--	19
20	117	1.89	2.97	0.59	0.87	3.66	NO	29	1
21	117	1.89	3.65	0.67	0.83	7.87	YES	--	1
22	117	1.89	4.48	0.76	0.80	7.32	YES	--	1
23	95	1.89	6.03	0.90	0.70	6.99	YES	--	10
24	85	1.89	6.03	0.90	0.70	6.50	NO	37	10

DATA POINT	a/h	L/D	P_{max} $\theta=0$	I_{max} $\theta=0$	Δt $\theta=0$	CENTER PT. DEFLECTION	FAILURE	% OF CIRCUM. DAMAGED	n
25	117	0.39	8.27	1.28	0.72	3.02	NO	25	30
26	85	0.39	23.44	2.33	0.40	2.64	NO	30	23
27	85	0.39	17.24	1.93	0.46	1.37	NO	34	25
28	95	0.89	5.52	1.03	0.91	0.86	NO	24	1
29	95	0.89	10.34	1.45	0.61	>11.43	YES	--	1
30	85	0.89	6.90	1.17	0.80	8.26	YES	--	1
31	85	0.89	8.27	1.31	0.72	1.60	NO	22	26
32	48	0.89	23.44	1.93	0.46	1.83	NO	24	11
33	95	1.89	5.52	1.03	0.91	>11.43	YES	32	1
34	95	1.89	5.52	1.03	0.91	5.56	YES	32	1
34	85	1.89	6.90	1.17	0.80	3.56	NO	32	1
36	85	1.89	8.27	1.31	0.72	7.21	YES	30	25
37	85	1.89	10.34	1.45	0.63	7.06	NO	33	25

 P_{max} ($\theta=0$) = Normally reflected pressure in megapascals (MPa), (1.0MPa=145psi) I_{max} ($\theta=0$) = Normally reflected impulse in megapascals-m sec (MPa-msec)

Center Pt. Deflection in centimeters (cm)

L/D = Length to diameter ratio

a/h = Radius to thickness ratio

n = Circumferential mode number

At = Positive pressure time phase in milliseconds (msec)

L/D values less than one have a decreasing mode number n for increasing thickness (see Table 2). This trend holds reasonably well for an L/D value of 0.89. For L/D values greater than one, however, the influence of change in thickness is less apparent.

Schuman [10] tested several sizes of cylinders subjected to blast loads but gave no response modes. His experimental results and those in Table 2 are generally in good agreement, but the analysis by Greenspon [11, 12] of Schuman's shells do not verify the results of Table 2. Another analysis [9] showed very good agreement for shells tested by the authors, but it predicted higher mode numbers than those in Table 2. The lack of correlation may be due to differences in the manner of loading and in calculated impulse values. Application of a modal type analysis [13] provided reasonable predictions of the final mode shape, but the method lacks appropriate criteria for predicting failure.

Determination of the load distribution for analysis is a major problem. A series of blast loads were imposed on a non-deforming cylinder using the loading methods described for the FAE and HE cylinder test. Experimental determination of the peak radial pressure distribution, as a function θ , approximated the expression

$$p_m = p_s + (p_r - p_s) (\cos \theta)^{1.8}$$

p_r and p_s are, respectively, the normal reflected pressure and the static pressure of a plane shock wave in air. Pressure measurements made by Lindberg [9] showed closer agreement to a $(\cos \theta)^2$ form. Time variations due to engulfment and decay were

$$p(\theta, t) = p_m [1 - (t - t_0)/\tau] \exp [-\alpha(t - t_0/\tau)]$$

where t_0 is the engulfment time based on shock wave speed, α is the approximate decay rate of the plane wave, and τ is the time of the positive pressure phase of the plane wave.

SUMMARY

Beams and plates respond similarly to blast loadings. Initially, at lower or mild blast loads, both beams and plates respond with a hinge type motion that traverses the entire width or length of the element;

failure or rebound then follows. For more severe loadings failure occurs early in the initial hinge motion. For intense loadings failure occurs as complete edge shear before any deflection takes place. For all plates and beams tested, failure occurred at the fixed ends or edges.

Cylindrical shell response to blast loading tends to be much less predictable than that for plates and beams and is complicated by a buckling phenomena that is dependent upon loading characteristics as well as the geometric and material properties of the cylindrical shell. In general, for a given cylindrical shell there exists a critical load which governs the response mode shape for the cylinder. Transverse blast loaded cylinders respond circumferentially in either a buckled or collapse mode coupled with a fundamental mode shape in the axial direction. For the cylinders tested, almost all of the damage occurred over only one-fourth of the circumference. The damage was centered around the leading edge of the cylinder; failure began as a crack at the fixed ends of the leading edge.

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THE FINITE ELEMENT APPLIED TO THE ANALYSIS OF MECHANISMS AND MACHINES

R.C. Winfrey*

Abstract - This review contains a survey of some approaches to the analysis of mechanisms. Complex models are described, as are various problems associated with the use of finite elements in such analyses.

Improvements in machinery frequently involve conflicting design goals -- for instance, both higher operating speeds and improved positioning accuracy. In this context, "high speed" is taken to mean any speed at which inertial forces are of sufficient magnitude that they cannot be ignored. If such inertial forces are ignored, stresses can increase because of resonance build-up, or failure can occur because of premature fatigue; at the least, overall performance is generally less than expected.

During the design process of a machine, it is convenient to use a simple mechanism as a model. A two- or three-dimensional model with simple finite elements can be used to analyze variable systems with such nonlinearities as damping, backlash, and clearances. The information obtained from studying a simple mechanism can be of great value in solving a complex problem.

This review is limited to a discussion of mechanisms. Various approaches to the analysis of mechanisms and the role of the finite element are described. Complex models containing clearances at the joints of the linkages are discussed, as well as problems associated with analyses of mechanisms and directions for future work.

Early attempts to include elastic effects in the analysis of mechanisms [1-7] were generally based on the slider-crank mechanism (Fig. 1) because of its simplicity. To further simplify the problem, elasticity was usually ignored in all members except the connecting rod (member 2 in the figure), and analog and/or digital computers were used to solve the derived equations of motion. More recent investigations [8] have made use of this simple model to study various effects of interest.

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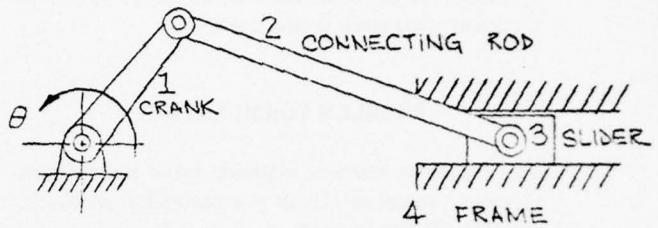


Figure 1. Slider-Crank Mechanism

The plane, four-bar mechanism (Fig. 2) was also given early consideration [9-10], and in 1969 the finite element method was used in general analyses of this and other mechanisms [11-15, 17]. Finite difference techniques were also applied to the analysis of mechanisms at about this time [16]; a unique method using an undulating elastica [7] was also introduced. The finite element method has become well established in engineering and it can be used to model two- and three-dimensional systems. A significant library of finite elements is now available in the literature.

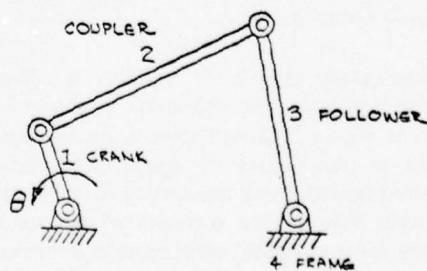


Figure 2. Four-Bar Mechanism

Early investigations were limited to analytical studies with little or no experimental verification. Among the first attempts at experimental verification were those by Alexander and Lawrence [18, 19], who used a slider crank model to confirm connecting rod resonances. They also verified the occurrence of significant fatigue stress reversals at five to ten times the driving frequency (crank speed).

PROBLEM FORMULATION

With the finite element method, either the stiffness approach, equation (1), or the flexibility approach, equation (2), can be used.

$$(F) = [k] (x) \quad (1)$$

$$(x) = [a] (F) \quad (2)$$

where $[a] = [k]^{-1}$

By definition, a mechanism allows rigid body deformations; therefore, the stiffness matrix $[k]$ is singular, and $[a]$ does not exist. The flexibility approach can be used by introducing artificial constraints, but the stiffness approach is more direct.

The simplest approach to modeling a four-bar mechanism is to use three classical beam elements and assume a rigid ground, as shown in Figure 3. It should be emphasized, however, that the rigid ground assumption is not made because of any limitations but for convenience. More coordinates could just as easily be added, and the frame included in the analysis, as shown in Figure 4. The added coordinates create more work for the computer, but not necessarily for the analyst.

Ten elastic link degrees of freedom, q_1 through q_{10} , and one rigid link degree of freedom, θ , are shown in Figure 3. Small deflections are usually assumed in calculations of elastic deflections. It has been shown [6] that accounting for large deflections adds little to the accuracy of the solution, primarily because elastic deflections in a functional machine are considered as second-order effects. Machine failure would occur long before the deflections increased appreciably. Because the elastic deflections are assumed to be small, they can thus be superimposed directly upon the rigid link mechanism.

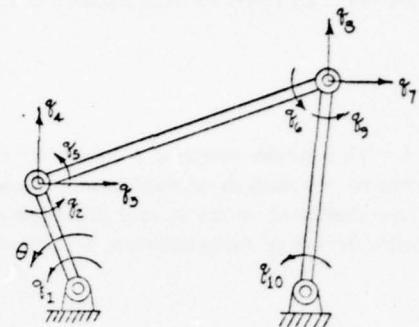


Figure 3. Four-Bar Mechanism - Rigid Base

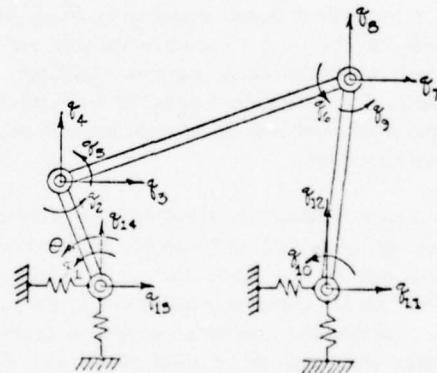


Figure 4. Four-Bar Mechanism - Elastic Base

A typical solution would include the following general steps:

- calculate the rigid body position, velocity, and acceleration of each link in the mechanism for a specific value of θ .
- use classical finite element methods to construct the dynamic equations of motion as if the mechanism were a stable structure.

$$[m](\ddot{q}) + [k](q) = (F) \quad (3)$$

- solve the equations of motion. Use as the initial conditions the results obtained as the final conditions of the previous elastic solution; superimpose the results upon the rigid body solution.
- $[m]$ and $[k]$ were obtained by assuming a fixed geometry but are actually functions of θ . Return to the first step and repeat when θ has changed enough -- perhaps one or two degrees, depending on the mechanism.

There are two major differences between finite element solutions for structures and for mechanisms. The obvious difference is that the geometry changes, so that $[m]$ and $[k]$ must be continuously recalculated - a significant task. The more subtle difference lies in the calculation of (F) in equation (3). Not only must (F) account for the usual external forces but also for the rigid body inertial forces. One approach [11, 14] used to obtain an expression for (F) is shown in equation 4.

$$(F) = (F)_{\text{external}} + (F)_{\text{inertial}} + (F)_{\text{relative}} \quad (4)$$

In the equation $(F)_{\text{external}}$ is the conventional set of externally applied loads; $(F)_{\text{inertial}}$ is somewhat analogous to a set of D'Alembert forces. These inertial forces arise from the rigid body accelerations of each link in the mechanism. The final term, $(F)_{\text{relative}}$, is like a Coriolis term. It arises because of the variable geometry and is a second order term, compared with $(F)_{\text{inertial}}$, for small elastic deflections. Thus, for the overall mechanism, $(F)_{\text{relative}}$ is essentially a second order effect and can be ignored. This is fortunate because its calculation can be cumbersome. Procedures for calculating $(F)_{\text{inertial}}$ can be found in the references [11].

METHODS OF SOLUTION

After equation (3) has been formulated, it must be solved to obtain the elastic deformations. The solution can be piecewise.

Modal Analysis

Modal analysis requires an eigenvalue routine for computing the eigenvalues and eigenvectors. The eigenvectors are then used to transform (q) into a set of modal coordinates (n) [24].

$$(q) = [\Phi](n)$$

where

$$[\Phi] = [(\phi)_1(\Phi)_2 \dots (\phi)_{10}] \quad (5)$$

and $(\phi)_j$ is the j th eigenvector.

Equation (5) is then applied to equation (3); the result is premultiplied by $[\Phi]^T$ to obtain the set of uncoupled differential equations shown in equation (8).

$$[\Phi]^T [m] [\Phi] (\ddot{n}) + [\Phi]^T [k] [\Phi] (n) = [\Phi]^T (F)$$

or,

$$[M]\ddot{(n)} + [K](n) = (N) \quad (8)$$

The solution to equation (8) for a step response is well known.

$$\begin{aligned} n_R(t) = & \frac{N_R}{M_{RR}W_R^2} (1 - \cos \omega_R t) + n_R(0) \cos \omega_R t \\ & + \frac{\dot{n}_R(0)}{\omega_R} \sin \omega_R t \end{aligned} \quad (9)$$

Equation (9) is used to find the system response during the short period of time, t that both the geometry and the forcing function are assumed to remain fixed.

Final values of (n) are transformed back to (q) with equation (5). The eigenvectors are an orthogonal set; if each vector is reduced to unit length, therefore,

$$[\Phi]^T = [\Phi]^{-1}$$

and the inverse transform is easily made. Thus, the

initial values at step $i+1$ are found from the previous final values at step i by

$$(n_0)_{i+1} = [\Phi]_{i+1}^T (q_f)_i$$

Modal damping can be included in equation (9), or some other classical form of damping can be introduced at an earlier stage.

One advantage of modal analysis is that relatively large steps can be taken as the mechanism rotates. A method for gaining more insight into how large a step can be taken under conditions of constant geometry has been described [14]. The major disadvantage to modal analysis is the time required to determine all the eigenvectors. A clever approach to reducing solution time -- supposedly by a factor of three -- was to estimate the rate of change in eigenvectors [15], thereby prolonging the calculation of new mass and stiffness properties. Another approach might be to use only one eigenvector, depending on the mechanism, because higher frequencies are usually less important than lower ones. The lowest frequency can be calculated quickly [24(pp 77, 78)].

Modal analysis is important in the dynamic analysis of linear elastic structures. For variable geometry problems, however, modal analysis has given way to direct, numerical integration techniques. One reason for this change is the long computation time required for modal analysis. Another is that research is being directed toward such highly nonlinear effects as clearance between members at their joints.

Numerical Integration

Numerical integration is an efficient way to solve both the older and the newer problems. The Runge-Kutta method [24] has been widely used; other schemes include the Newmark method [20] and the Wilson- θ method [21]. The Newmark method is simple because a linearly varying acceleration is assumed. The Wilson- θ method is somewhat more complex but can be shown to be unconditionally stable.

The major difficulty in using numerical integration to analyze mechanisms is that the links are essentially beam members. Frequencies associated with axial motion are therefore usually several orders of magnitude higher than frequencies associated with bending motion. Thus, even though axial motion is of little

concern, it must be accounted for in the determination of an integration time step. The problem can be avoided by eliminating axial, elastic degrees of freedom, but this must be done with care to avoid interference with the rigid body axial motion and with the bending modes of adjacent links.

ADVANCED TOPICS AND FUTURE TRENDS

One of the most exciting and challenging topics to come out of the application of the finite element method to mechanisms has been the study of impact and the effect of clearance at the joints between links. A great deal of effort is also being directed at gearing applications. The impact between two bodies has been studied for some time, of course, but the application of the finite element method is new.

The coefficient of restitution adequately accounts for the loss of energy and general behavior of such simple systems as a bouncing ball. Its main failing is that motion before impact is related to motion after impact; *what happens during impact* is ignored. A pin in a practical mechanism joint will have a close fit with its bearing; the time of impact is thus a significant part of the total time. A better model of impact is needed.

The impact damper is an example of a simple mechanism with joint clearance. It consists of a box, or enclosure, containing a ball that is allowed to roll back and forth through a small, carefully controlled distance. The idea is not new [25] and has in fact been studied for the past ten years [26-30]. Dubowsky proposed a model for an impact pair [31] and later made experimental studies [32]. The clever experiment was a quasi-inversion of the box-ball configuration; both the acceleration of the freely moving mass (ball) and the box could be directly measured. More recently, others have reported experimental and theoretical work on similar configurations [33-35]. Obviously, much is to be learned from this simple device.

The finite element method was first applied to simple, one-dimensional impact models such as the cam/follower mechanism [30, 37, 38]. One-dimensional impact implies that impact occurs along a single line -- as opposed to the much more complex two-dimensional impact situation which occurs, for

example, between a pin and its mating hole. The study of two-dimensional impact was recently applied to rigid link mechanisms [39, 40]; the more difficult concept of elasticity has also been included in the links [41].

Modeling of large deflections has not received much attention. An undulating elastica [7] and the finite element method in a nonlinear, piecewise fashion [42] have been used.

The finite element models described above are complex, and, of course as the complexity of both the model and its nonlinear elements increases, so must the computer time required to solve the equations of motion. For the conventional structural analysis of linear systems, a large problem may have from 1,000 to 50,000 degrees of freedom or more. Even though the simple mechanisms discussed in this review are typically modeled with 10 or 15 degrees of freedom, computer times tend to be excessive for repetitive parameter studies. A few attempts have been made to reduce computer time with simplified models [14, 43] and more efficient coding [15, 44], but a method for significantly reducing solution times without affecting accuracy has not yet been developed.

Another problem facing the analyst using complex models is the proper display of the voluminous data produced by the computer. It is difficult enough to understand what is actually happening to a machine as a pin "rattles" around in a bearing. When a number of joints are rattling at the same time, it is almost impossible to determine if the vibrations can be reduced -- by changing the size of a clearance or by slightly adjusting the geometry or mass distribution. Yet, these are the types of solutions that must be sought in order to build faster, more precise machinery.

CONCLUDING REMARKS

The application of the finite element method to the analysis of mechanisms has been a challenging task during the 1970s. Considerable insight has accumulated with regard to techniques for efficiently analyzing sets of highly nonlinear equations. The results of recent studies will be manifest as a capability to more accurately predict the behavior of new

machines before they are built, and will also serve as a guide for the trouble-shooting of existing machines.

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ANNUAL ARTICLE INDEX

FEATURE ARTICLES

	ISSUE	PAGES
Done, G.T.S. Vibration of Helicopters	1	5-13
Vance, J.M. Absorbers and Isolators for Torsional Vibration	2	3-6
Matsuzaki, Y. A Review of Shock Response Spectrum	3	3-12
Birchak, J.R. Damping Capacity of Structural Materials	4	3-11
Rieger, N.F. Rotor-Bearing Dynamics: State-of-the-Art 1976	5	5-14
Berkof, R.S., Lowen, G.G., and Tepper, F.R. Balancing of Linkages	6	3-10
Nelson, F.C. Techniques for the Design of Highly Damped Structures	7	3-11
Traexler, J.F. Turbomachinery Vibration	8	3-10
Firth, D. Acoustic Vibration of Structures in Liquids	9	3-7
Tustin, W. A Comparison of Techniques and Equipment for Generating Vibration	10	3-10
Craig, R.R., Jr. Methods of Component Mode Synthesis	11	3-10
Oleson, M.W. and Belsheim, R.O. Shipboard Shock Environment and Its Measurement	12	3-12

LITERATURE REVIEWS

	ISSUE	PAGES
Mitchell, W.S. Shock and Vibration Instrumentation: Accelerometers	1	15-18
Derby, T.F. Computer Programs: Shock and Vibration Isolation	1	19-26
Gibson, R.F. and Plunkett, R. Dynamic Stiffness and Damping of Fiber-Reinforced Composite Materials	2	9-17
Krishna Murty, A.V. Finite Element Modeling of Natural Vibration Problems	2	19-37
Rao, J.S. Turbine Blading Excitation and Vibration	3	15-22
Chen, F.Y. A Review of the Literature on the Dynamics of Cam Mechanisms	3	23-36
Jensen, J.J. and Madsen, N.FI. A Review of Ship Hull Vibration. Part 1: Mathematical Models	4	13-22
Chen, L.H. and Pierucci, M. Underwater Fluid-Structure Interaction. Part I: Introduction and Scope	4	23-24
Chen, L.H. and Pierucci, M. Underwater Fluid-Structure Interaction. Part II: Mechanically-Applied Forces	5	17-23
Jensen, J.J. and Madsen, N.FI. A Review of Ship Hull Vibration. Part II: Modeling Physical Phenomena	5	25-38
Chen, L.H. and Pierucci, M. Underwater Fluid-Structure Interaction. Part III: Acoustically-Applied Forces	6	13-17
Jensen, J.J. and Madsen, N.FI. A Review of Ship Hull Vibration. Part III: Methods of Solution	6	19-27
Jensen, J.J. and Madsen, N.FI. A Review of Ship Hull Vibration. Part IV: Comparison of Beam Models	7	13-28
Chen, L.H. and Pierucci, M. Underwater Fluid-Structure Interaction. Part IV: Hydrodynamically Applied Forces (Moving Medium)	7	29-37
Ward, H.S. The Characteristics of Dynamic Loads and Response of Buildings	8	13-20

LITERATURE REVIEWS (CONTINUED)

Munjal, M.L. Exhaust Noise and Its Control - A Review	8	21-32
Krajcinovic, D. Some Transient Problems of Structures Interacting with Fluid	9	9-16
Wagner, H. Beam Vibrations - A Review	9	17-24
Leissa, A.W. Recent Research in Plate Vibrations: Classical Theory	10	13-24
Chen, S.-S. Flow-Induced Vibrations of Circular Cylindrical Structures. Part I. Stationary Fluids and Parallel Flow	10	25-38
Prause, R.H. Dynamic Modeling of Pressure Vessels and Piping Systems	11	13-20
Chen, S.-S. Flow-Induced Vibrations of Circular Cylindrical Structures. Part II: Cross-Flow Considerations	11	21-27
Ross, C.A., Strickland, W.S., and Sierakowski, R.L. Response and Failure of Simple Structural Elements Subjected to Blast Loadings	12	15-26
Winfrey, R.C. The Finite Element Applied to the Analysis of Mechanisms and Machines	12	27-33

BOOK REVIEWS

HANDBOOK OF PYROTECHNICS

K.O. Brauer

Chemical Publishing Co., Inc. (1974)

The Handbook of Pyrotechnics is a fascinating book that seems to fulfill the expressed intent of the author:

"It is the purpose of this handbook to provide useful data and information about theory and practical application of pyrotechnics for engineers, designers, technicians and students."

No previous knowledge of the subject is assumed and the material is presented in an almost "popular" way. Thus it can either be scanned rapidly for basic ideas or individual mechanisms can be studied in more detail.

A quotation from the author's introduction outlines the contents:

"The contents of this handbook are divided into six parts: Explosive Materials, Explosive-Actuated Devices, Pyrotechnic Systems, Reliability and Testing, Explosive Production Methods, and Appendix.

The handbook contains numerous charts, graphs, and illustrations as useful aids. Theory, data, and practical applications are explained in detail. Valuable new information is presented in this handbook, as for example data about the effects of extreme environmental conditions on pyrotechnic materials and devices, hints and data for qualification testing, hints for the design and application of pyrotechnic systems, and data for the application of explosive methods in manufacturing processes.

It is recommended to use this handbook together with the book Military and Civilian Pyrotechnics by Dr. Herbert Ellern, published by the Chemical Publishing Company, which contains more detailed information about the properties, and produc-

tion of pyrotechnic materials and an extensive manufacturing formulary."

The book is filled photographs, sectioned drawings, schematics, sequence diagrams, tables, and graphs that describe the operation and construction of specific devices and systems. The information can be very useful for a designer attempting to solve a problem. It is not detailed enough for him to complete a design solution but is a good source of possible approaches.

The book contains a wealth of descriptive material on spacecraft systems and a lesser amount on aircraft and missile systems. Manufacturing uses are covered briefly but well. The book contains a reasonable glossary and 95 references, most of which are from open literature periodicals and books. Although credits are given for the many photographs and diagrams, company literature, which must have provided sources, is not mentioned. Some way for readers to contact producers and developers would be a useful addition to the book.

Although readers of the DIGEST might find this handbook interesting and useful, they will no doubt realize that one subject has not been included: that of the shocks produced by the various pyrotechnic devices. DIGEST readers would find such information useful -- even rudimentary typical descriptions. A classification of devices according to shocks produced would also be a helpful tool for designers.

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VIBRATION OF BEARINGS (Vibratsiya Podshipnikov)

K. Ragulskis, A. Jurkauskas, V. Astupenas, A. Vitkute, and A. Kulvec

Leidykla Mintis, Vilnius, Lithuania (1974)

(In Russian)

The vibration of bearings became important with the invention of the wheel. No one knows exactly when it was invented or by whom. However, archaeologists have found evidence of the existence of wheels in graves that date from 5,500 years ago. A large body of literature has accumulated pertaining to the vibration of bearings; it is scattered in many publications throughout the world. In general each publication is concerned with study of only a part of the "total problem of vibration of bearings."

The authors of this book are associated with the Kaunas Polytechnic Institute in Lithuania. They were assisted by A.B. Palionis, R.P. Atstunenene, R.V. Kanapenas, V.I. Zdanavichyus, V.N. Augutis, and I.R. Zhitkevichyus, all of whom are also associated with the Kaunas Institute. Their book is a welcome addition to the literature of vibration of bearings, especially because it treats the "total problem of vibration of bearings." The book provides an excellent summary of the current state-of-the-art of vibration of bearings in Eastern European countries.

The book is concerned with analytical and experimental investigation of bearings and with the design of bearings and bearing units. It contains the following chapters:

1. Analytical determination of rotational resisting moments.
2. Determination of the elastic and damping characteristics of oscillating bearings.
3. Analysis of radial vibration of bearings and bearing units.
4. Methods and equipment for measurement of dynamic characteristics of bearings.
5. Errors in measurement of rotational resisting moments and means for reducing them.
6. Method for statistical treatment of experimental results obtained from investigation of the dynamics of precision bearings.
7. Experimental investigation of the dynamic characteristics of precision bearings and their units.

8. Methods and schemes to reduce the rotational resisting moments and the vibration of bearings.

The analytical determination of rotational resisting moments is based on theory that has evolved during the past 15 years. The theory assumes that this total moment is composed of a sum of eight components, all of which are multiplied by a single corrective coefficient to account for factors that cannot be accounted for analytically. Thus, a complete analytical theory for rotational resisting moments of bearings that properly accounts for all relevant factors remains to be developed.

Of especial interest in the book are the experimental data on rotational resisting moments versus rotational speed for various elevated temperatures. The last chapter in the book will also be of interest to designers who sometimes state "Don't bother me with the theory and the experimental results -- just tell me in plain English how I can reduce the rotational resisting moments and the vibration of bearings."

References in this book by number were: Eastern European countries (470), German (61), English (33), and Italian (1). In some chapters the text refers to references that are not listed in the bibliography following the chapter. A number of typographical errors can be found in most of the bibliographies following each chapter. Only 1,000 copies of the book have been printed.

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**NONLINEAR AND LINEAR TRANSIENT
DEFORMATION WAVES IN THERMO-
ELASTIC AND ELASTIC BODIES**
**(Nelineinyye i lineinyye perekhodnye volnovyye
protsessy deformatsii termouprugikh i
uprugikh tel)**

U.K. Nigul and Yu. K. Engel'brekht
Tallin, Akademiya Nauk Estonskoi SSR
Institut Kibernetiki (1972)

This monograph presents a systematic treatment of transient wave processes in continuous media from a differential equation approach. A rather extensive bibliography is included and takes up almost two-fifths of the entire manuscript. Aside from summary treatment of works by other authors, it is primarily a reiteration of the work of the first author.

Although a pretense in rigor was attempted, nothing more than academic solution was advanced. In particular, the authors made only a passing remark in the introduction of artificial damping in the solution of a moving shock front (Section 7.2, p. 84) and expressed their doubts as to its validity, apparently being unaware of the rather important contribution by von Neumann using artificial damping to enable rapid and efficient numerical integration of wave problems with high gradients. Finite-element treatment, likewise, received only scanty passing mention. Boley's treatment of Timoshenko-type beams, using solutions for separate regions (1955, 1956), was cited as a starting point for some new development in USSR by Slepian (Section 9.2, p. 91). However, no description of Slepian's work was included in the monograph, consequently tending to leave this reviewer completely in the dark as to what is the real improvement in the technique.

The monograph is divided into three parts. Part 1 examines the general equations governing thermoelasticity with due consideration of nonlinear equations of thermal conductivity. It is shown that for typical structural materials, the effects due to geometrical nonlinearities are of the same order of magnitude as those due to physical nonlinearities. In part 2, methods of analysis of transient wave processes caused by mechanical inputs are examined and classified. In part 3, one-dimensional example is treated to exhibit nonlinear and thermal effects.

This reviewer considers that this monograph, although lacking in some degree a complete survey of contemporary techniques, nevertheless provides a clear and concise systematic identification of the problems involved; i.e., it is a good primer to the subject, but not very useful for anything else.

C.C. Wan, USA
Courtesy of Applied Mechanics Reviews

ROTOR VIBRATIONS AND BALANCING
(Kolebaniya i uravnoveshchivanie rotorov)
Izdatelstvo "Nauka", Moscow (1973)
(\$2.50)

The book consists of 18 papers on some recent problems in vibrations of high-speed rotors and their balancing.

The first six consider vibrations of flexible rotors and deal with: vertical rotors under gravitational forces, (M. F. Zeitman); excitation of counterprecession, (G. I. Anikieiev); influence of disk dimensions on natural frequencies of excited vibrations of step rotors under concentrated forces, (A. A. Gusarov); dynamic deflections of eccentric rotors, (N. G. Samarow), and determination of optimal parameters, (M. F. Zeitman and R. B. Statnikov).

Papers dealing with balancing problems can be grouped as those concerned with: (1) the influence of balancing weight distribution, (N. G. Samarov and L.N. Kudriaszew), their flexible mounting insensitive balancing speeds, (A. A. Gusarov); (2) applications of amplitude-phase characteristics to rotor balancing (L. Ia. Banach, M. D. Piermikov, and L. N. Shatalov), (A. A. Gurarov and L. N. Ghatalov); (3) automatic balancing, (W. I. Susanin), (M. D. Genkin and others). Other special and more general problems of balancing and measurement are also considered by M. E. Levit, A. I. Maximienko, Iu. A. Samsaev, K. W. Frolov, T. P. Kozlanikov, and Iu. A. Pietrov.

The general introduction is given by A. A. Gusarov.

Z. Parszewski, Poland
Courtesy of Applied Mechanics Reviews

SHORT COURSES

JANUARY

EARTHQUAKE SIMULATION AND RESPONSE

Dates: January 9 - 13, 1978

Place: Long Island, New York

Objective: Safe shut-down of a nuclear power generating station following an earthquake will be the main topic of this course, to be held at the facilities of Dayton T. Brown, Inc., Bohemia, Long Island, New York -- one of the few laboratories in the world capable of earthquake simulation. This course, aimed at test and quality engineers, will stress interpreting standards and specifications and conducting tests (including the proper mounting of test specimens).

Contact: Wayne Tustin, Tustin Institute of Technology, 22 East Los Olivos St., Santa Barbara, CA 93105 (805) 953-1124.

MAINTAINABILITY ENGINEERING

Dates: January 9 - 13, 1978

Place: UCLA Extension

Objective: This course is designed to help participants to determine the following: the distribution of times-to-repair components and times-to-restore equipment, the equipment mean-time-to-restore, the mean man-hours needed to restore, the optimum preventive maintenance schedules for minimum total corrective and preventive maintenance cost, spare parts requirements with a specified assurance and their optimization, the reliability, maintainability and availability (both instantaneous and steady state) of maintained equipment and systems, interpret and use MIL/STB-471 and MIL/DBK-472. The course is intended for those involved in the conception, design, operation and maintenance of any equipment in today's mechanical society. A Bachelor's degree in engineering, mathematics, or equivalent is required.

Contact: Continuing Education in Engineering and Mathematics, Short Courses, 6266 Boelter Hall, UCLA Extension, Los Angeles, CA 90024 (213) 825-3344 or 825-1295.

FEBRUARY

VIBRATION SURVIVABILITY

Dates: February 6 - 10, 1978

Place: Fullerton, California

Objective: This course, which will be held at the facilities of Hughes Aircraft, Malvern at Gilbert Sts., Fullerton, Calif., is designed to provide basic education in resonance and fragility phenomena, in environmental vibration and shock measurement and analysis, also in vibration and shock environmental testing to prove survivability. This course will concentrate upon techniques and equipments rather than upon mathematics and theory.

Contact: Wayne Tustin, Tustin Institute of Technology, 22 E. Los Olivos St., Santa Barbara, CA, 93105 (805) 963-1124.

MACHINERY VIBRATION MONITORING AND ANALYSIS SEMINAR

Dates: February 13, 14, & 15, 1978

Place: Houston, Texas

Objective: This seminar will be devoted to the understanding and application of vibration technology to machinery vibration monitoring and analysis. Basic and advanced techniques with illustrative case histories and demonstrations will be discussed by industrial experts and consultants. Topics to be covered in the seminar include preventive maintenance, measurements, analysis, data recording and reduction, computer monitoring, acoustic techniques, misalignment effects, balancing, turbomachinery blading, bearing fault diagnosis, torsional vibration problems and corrections, and trend analysis. An instrumentation show will be held in conjunction with this seminar.

Contact: Dr. R. L. Eshleman, Vibration Institute, Suite 206, 101 W. 55th St., Clarendon Hills, IL, 60514 (312) 654-2254.

NEWS BRIEFS

 news on current
and Future Shock and
Vibration activities and events

CALL FOR PAPERS 1979 Fifth World Congress on the Theory of Machines and Mechanisms

The Fifth World Congress on the Theory of Machines and Mechanisms, to be held at Concordia University, Montreal, Canada, during July 8 - 13, 1979, will be a forum to discuss all aspects of problems related to the theory of machines and mechanisms and applied problems.

Delegates from all over the world are expected and papers are solicited in the areas of kinematic analysis and synthesis; dynamics of machines and mechanisms; gearing and transmissions; preventive maintenance and reliability control; rotor-dynamics; vibrations and noise in machines; biomechanisms; technology transfer; robots, manipulators and man-machine systems; computer-aided design and optimization; pneumatics, hydraulics and electro-dynamics; industrial applications for special machines and mechanisms; experimental and teaching methods.

For further information, please contact:

Dr. Seshadri Sankar
Papers Review and Program Chairman
IFTOMM Congress
Dept. of Mechanical Engineering
Concordia University
1455 de Maisonneuve W.
Montreal, Canada H3G 1M8
Telephone (514) 879-5839

INSTITUTE OF ENVIRONMENTAL SCIENCES Shock and Vibration Test Problems Subcommittee

The Test Problems Subcommittee of the IES Shock and Vibration Committee under the chairmanship of Wayne Tustin, Tustin Institute of Technology, will compile a list and describe the most critical problems

in shock and vibration testing practice. The subcommittee expects the list will be complete by December 31, 1977 and published in draft shortly thereafter.

The subcommittee is soliciting input and assistance.

Subcommittee members are:

Wayne Tustin, Chairman
John Losse, Delco
Dick Shelby, Hughes Aircraft
Darrell Dickey, Raytheon

Contact Wayne Tustin at Tustin Institute of Technology, 22 E. Olivos St., Santa Barbara, CA 93105 (805) 963-1124.

SECOND WORLD CONGRESS ON FINITE ELEMENT METHODS Bournemouth, Dorset, England 23rd to 27th October, 1978

The Second World Congress on Finite Element Methods is to be held at the Royal Bath Hotel, Bournemouth, Dorset, England, 23rd to 27th October, 1978. A Finite Element Method Exhibition will also be held at the same event. The theme of the Congress is finite element methods in the commercial environment and Professor R. H. Gallagher, Cornell University, will deliver the main invited lecture.

For further information, please contact:

Dr. John Robinson
Robinson and Associates
Horton Road, Woodlands, Wimborne
Dorset BH21 6NB England

ABSTRACT CATEGORIES

ANALYSIS AND DESIGN

Analogs and Analog Computation
Analytical Methods
Dynamic Programming
Impedance Methods
Integral Transforms
Nonlinear Analysis
Numerical Analysis
Optimization Techniques
Perturbation Methods
Stability Analysis
Statistical Methods
Variational Methods
Finite Element Modeling
Modeling
Digital Simulation
Parameter identification
Design Information
Design Techniques
Criteria, Standards, and Specifications
Surveys and Bibliographies
Tutorial
Modal Analysis and Synthesis

COMPUTER PROGRAMS

General
Natural Frequency
Random Response
Stability
Steady State Response
Transient Response

ENVIRONMENTS

Acoustic
Periodic
Random
Seismic
Shock
General Weapon
Transportation

PHENOMENOLOGY

Composite
Damping
Elastic
Fatigue
Fluid
Inelastic
Soil
Thermoelastic
Viscoelastic

EXPERIMENTATION

Balancing
Data Reduction
Diagnostics
Equipment
Experiment Design
Facilities
Instrumentation
Procedures
Scaling and Modeling
Simulators
Specifications
Techniques
Holography

COMPONENTS

Absorbers
Shafts
Beams, Strings, Rods, Bars
Bearings
Blades
Columns
Controls
Cylinders
Ducts
Frames, Arches
Gears
Isolators
Linkages
Mechanical
Membranes, Films, and Webs

Panels
Pipes and Tubes
Plates and Shells
Rings
Springs
Structural
Tires

SYSTEMS

Absorber
Acoustic Isolation
Noise Reduction
Active Isolation
Aircraft
Artillery
Bioengineering
Bridges
Building
Cabinets
Construction
Electrical
Foundations and Earth
Helicopters
Human
Isolation
Material Handling
Mechanical
Metal Working and Forming
Off-Road Vehicles
Optical
Package
Pressure Vessels
Pumps, Turbines, Fans, Compressors
Rail
Reactors
Reciprocating Machine
Road
Rotors
Satellite
Self-Excited
Ship
Spacecraft
Structural
Transmissions
Turbomachinery
Useful Application

ABSTRACTS FROM THE CURRENT LITERATURE

Copies of articles abstracted in the DIGEST are not available from the SVIC or the Vibration Institute (except those generated by either organization). Inquiries should be directed to library resources. Government reports can be obtained from the National Technical Information Service, Springfield, VA 22151, by citing the AD-, PB-, or N- number. Doctoral dissertations are available from University Microfilms (UM), 313 N. Fir St., Ann Arbor, MI; U.S. Patents from the Commissioner of Patents, Washington, D.C. 20231. Addresses following the authors' names in the citation refer only to the first author. The list of periodicals scanned by this journal is printed in issues 1, 6, and 12.

ABSTRACT CONTENTS

ANALYSIS AND DESIGN	44	Shock	53	Linkages	62
Analytical Methods	44	General Weapon	53	Pipes and Tubes	63
Impedance Methods.	44	Transportation	53	Plates and Shells	64
Nonlinear Analysis.	44			Structural	67
Numerical Analysis	44				
Optimization Techniques	44	Damping	53	SYSTEMS	67
Stability Analysis	45	Elastic.	54	Absorber	67
Statistical Methods	45	Fluid.	55	Noise Reduction	68
Finite Element Modeling.	45	Soil.	55	Aircraft.	69
Parameter Identification	45	Viscoelastic	56	Bridges	71
Design Techniques.	46			Building.	71
Criteria, Standards, and				Foundations and Earth.	72
Specifications	46			Helicopters.	72
Surveys and Bibliographies .	46	Diagnostics.	56	Human	73
Modal Analysis		Facilities	57	Isolation	73
and Synthesis	48	Instrumentation	57	Mechanical.	73
		Techniques.	57	Metal Working	
				and Forming.	73
COMPUTER PROGRAMS	48			Pumps, Turbines, Fans,	
General	48	Absorbers	58	Compressors.	74
ENVIRONMENTS.	50	Beams, Strings, Rods, Bars	58	Rail	74
Acoustic	50	Bearings.	59	Reactors	75
Periodic.	51	Blades.	60	Road.	75
Random	51	Ducts	60	Rotors.	76
Seismic	52	Frames, Arches.	61	Spacecraft	77
		Gears	61	Turbomachinery	77

ANALYSIS AND DESIGN

ANALYTICAL METHODS

77-2044

The Effect of Delay on the Behavior of a Nonlinear Vibration System (Über den Einfluss von Totzeiten auf nichtlineare Schwingungssysteme)

J.A. Mitropolskij

Math. Inst. of the Academy of Sciences; Uliza Repina 3, 25 260 1 Kiew, USSR, Ing. Arch., 45 (5/6), pp 387-392 (1976) 8 refs
(In German)

Key Words: Nonlinear systems, Vibrating structures

The effect of delay on the behavior of a nonlinear oscillating system is investigated. Qualitative analysis has been carried out for some practically important problems and the influence of delay effects on the oscillation properties; namely, type of oscillation, stability, nature and intensity of damping have been examined.

IMPEDANCE METHODS

(See No. 2080)

NONLINEAR ANALYSIS

77-2045

Parametric Vibration of a Non-Linear System

A. Tondl

National Research Inst. for Machine Design, 25097 Praha 9 - Bechovice, CSSR, Ing. Arch., 45 (5/6), pp 317-324 (1976) 9 figs, 3 refs

Key Words: Nonlinear systems, Single degree of freedom systems, Parametric response

An analysis is presented of a non-linear system with one degree of freedom, in which the restoring force is expressed by the product of a periodic function of time and a non-linear function of deflection. In such a system there can occur not only the expected parametric resonances of the order n ($n = 1, 2, \dots$) but resonances of the order $1/N$ ($N = 2, 3, \dots$) as well.

NUMERICAL ANALYSIS

77-2046

Analysis and Design of Numerical Integration Methods in Structural Dynamics

H.M Hilber

Ph.D. Thesis, Univ. of California, Berkeley, 111 pp (1976)
UM 77-15,718

Key Words: Numerical analysis, Dynamic structural analysis

The objective of this work is to develop one-step methods for the integration of the equations of structural dynamics which are unconditionally stable, have an order of accuracy not less than two, and possess numerical dissipation which can be controlled by a parameter other than the time step size. In particular, no numerical dissipation is included. Four new families of algorithms are discussed from this point of view, and compared with algorithms, such as the Newmark, Wilson and Houbolt methods, which are commonly used in structural dynamics and do not achieve these requirements.

77-2047

A Splitting Method for Computing Coupled Hydrodynamic and Structural Response

J.E. Ash

Argonne National Lab., Argonne, IL 60439, Appl. Math. Modeling, 1 (6), pp 333-338 (Sept 1977) 4 figs, 5 refs

Key Words: Numerical analysis, Nuclear reactor containment, Underwater explosion, Hydrodynamic excitation

A numerical method is developed for application to unsteady fluid dynamics problems. In particular to the mechanics following a sudden release of high energy. Solution of the initial compressible flow phase provides input to a power-series method for the incompressible fluid motions. The system is split into spatial and time domains which lead to the convergent computation of a sequence of elliptic equations. Two sample problems are solved.

OPTIMIZATION TECHNIQUES

77-2048

Reliability-Based Optimization for Dynamic Loads

J.W. Davidson, L.P. Felton, and G.C. Hart

Ameron, South Gate, CA., ASCE J. Struc. Div.,

103 (ST10), pp 2021-2035 (Oct 1977)

Key Words: Minimum weight design, Shock response, Probability theory

A general formulation is presented for weight optimization of indeterminate structures subject to transient dynamic loads and reliability constraints. Two distinct methods of structural analysis are examined and compared for use in the optimization algorithm: Numerical integration of equations of motion and shock spectra. Details of the essential computation of standard derivation of response quantities associated with each analysis technique are also examined. The formulations are illustrated by design examples of a rigid frame subjected to an acceleration impulse applied to its base.

77-2049

Structural Properties of Linear Dynamic Systems: Application to Optimal Control and Filtering

O.L. Mercier

Office National d'Etudes et de Recherches Aero-spatiales, Paris, France, Rept. No. ONERA-NT-1977-4, FR-ISSN-0078-3781, 26 pp (Mar 1977) refs (In French)
N77-25859

Key Words: Optimum control theory, Dynamic systems

The major results concerning the modern concepts of controllability, observability, reconstructability, stability, stabilizability, and detectability of linear dynamic systems are presented. These concepts, developed during the 1960 to 1972 period, are of prime importance for the control of dynamic systems, especially to design feedback controls and to synthesize the filters, state reconstructors, and observers usually associated with these controls.

STABILITY ANALYSIS

77-2050

Energy Expressions as Stability Criteria in Linear Differential Equations with Periodic Coefficients (Energieausdrücke als Stabilitätskriterien bei linearen Differentialgleichungen mit Periodischen Koeffizienten)

E. Brommundt

Mechanik-Zentrum, Lehrstuhl A für Mechanik, Technische Universität Braunschweig, Postfach 3329, D-3300 Braunschweig, Federal Rep. of Germany, Ing. Arch., 45 (5/6), pp 325-330 (1976) 2 refs (In German)

Key Words: Stability, Turbomachinery, Perturbation technique

Starting from the principle of virtual work the stability of the trivial solution is investigated by means of a perturbation technique. The stability conditions have the form of energy expressions which, in general, cannot be interpreted as energy flows.

STATISTICAL METHODS

(See Nos. 2087, 2125)

FINITE ELEMENT MODELING

(Also see No. 2140)

77-2051

Solutions to Initial Value Problems by Use of Finite Elements -- Unconstrained Variational Formulations

J.J. Wu

Benet Weapons Lab., Watervliet Arsenal, Watervliet, NY 12189, J. Sound Vib., 53 (3), pp 341-356 (Aug 8, 1977) 2 figs, 5 tables, 15 refs

Key Words: Boundary value problems, Finite element technique, Forced vibration

This paper presents a variational formulation which treats initial value problems and boundary problems in a unified manner. The basic ingredients of this theory are adjoint variable and unconstrained variations. It is an extension of the finite element unconstrained variational formulation used previously in solving several non-conservative stability problems. The technique which makes this extension possible is described. This formulation thus enables one to adapt such numerical techniques as the finite element method, which has had great success and popularity for solution of boundary value problems, for solutions of initial value problems as well.

PARAMETER IDENTIFICATION

(Also see No. 2177)

77-2052

Maximum Likelihood Parameter Identification of Linear Dynamic Systems

F. Chen

Ph.D. Thesis, Northeastern Univ., 114 pp (1977)
UM 77-17,784

Key Words: Linear systems, Parameter identification

This dissertation develops and compares two maximum-likelihood methods for parameter estimation. It includes: Formulation and comparison of the performance criteria for two maximum-likelihood methods, denoted as ML1 and ML2, Derivation of an equivalent ML2 criterion and a numerical procedure to provide the estimation of the state and the unknown parameter vector separately, Investigation and comparison of the estimation properties of the ML1 and ML2 methods with numerical examples included.

physical interpretation and the subsequent exploitation of the mathematical models. Various applications of the technique are also described, and the future development is envisaged.

DESIGN TECHNIQUES

(See Nos. 2068, 2069)

CRITERIA, STANDARDS, AND SPECIFICATIONS

(See No. 2154)

77-2053

Correction of the Theoretical Model of an Elasto-mechanical System by Means of Measured Forced Vibrations (Die Korrektur des Rechenmodells eines elastomechanischen Systems mittels gemessener erzwungener Schwingungen)

H.G. Natke

Lehrstuhl für Schwingungs- und Messkunde und Curt-Risch-Institut, Technische Universität Hannover, Callinstr. 32, D-3000 Hannover, Federal Rep. of Germany, Ing. Arch., 46 (3), pp 169-184 (1977) (In German)

Key Words: Mathematical models, Parameter identification

The system analysis of elastomechanical systems results in a theoretical model as an approximation of the real structure. The system identification leads to the incomplete experimental model. The quality criterion applied to the theoretical model may be the accordance of the eigencharacteristics of the theoretical model with the eigencharacteristics of the experimental model or the accordance of their frequency responses.

77-2054

Dynamic Data System: A New Modeling Approach

S.M. Wu

Dept. of Mech. Engrg., Univ. of Wisconsin, Madison, WI, J. Engr. Indus., Trans. ASME, 99 (3), pp 708-714 (Aug 1977) 4 figs, 45 refs

Key Words: Mathematical models, Parameter identification

The dynamic data system is a modeling technique that uses dynamic data in the form of a time series to develop physically meaningful stochastic difference/differential equations. The general mathematical formulation and background of the dynamic data system methodology are given, and the modeling procedure evolved in this approach is illustrated by an example pertaining to neutron flux data. An example of a machine tool system analysis is presented to show the

SURVEYS AND BIBLIOGRAPHIES

(Also see No. 2104)

77-2055

Acoustic Holography (Citations from the Engineering Index Data Base)

W.E. Reed

National Technical Information Service, Springfield, VA., Rept. No. NTIS/PS-77/0579/1GA, 218 pp (July 1977)

Key Words: Acoustic holography, Bibliography

Worldwide research on acoustic holography is covered. Theory, uses, equipment design, and imaging techniques are presented. Most of the studies are general and not applied to a specific use of acoustic holography. However, there are citations which do discuss its use in medicine, nuclear reactors, and nondestructive testing. (This updated bibliography contains 211 abstracts, 50 of which are new entries to the previous edition.)

77-2056

Acoustic Holography (Citations from the NTIS Data Base)

W.E. Reed

National Technical Information Service, Springfield, VA., Rept. No. NTIS/PS-77/0578/3GA, 130 pp (July 1977)

Key Words: Acoustic holography, Bibliography

All aspects of acoustic holography are covered in this bibliography of Federally-funded research. Theory, equipment design, uses, and imaging techniques are presented. The applications include underwater and underground object

locating, structural geology and tectonics, sonar imaging, non-destructive testing, antenna radiation patterns, nuclear reactor inspection, remote sensing, and use in medical examinations. (This updated bibliography contains 125 abstracts, 23 of which are new entries to the previous edition.)

77-2057

Environmental Pollution: Noise Pollution - Sonic Boom

Defense Documentation Center, Alexandria, VA., Rept. No. DDC/BIB-77/06, 201 pp (June 1977) AD-A041 400/3GA

Key Words: Sonic boom, Bibliographies

This bibliography contains citations of studies and analyses covering a wide range of the parameter of sonic boom and noise pollution, as well as damages caused by it. Corporate Author-Monitoring Agency, Subject, Title and Personal Author are provided.

77-2058

The Characteristics of Dynamic Loads and Response of Buildings

H.S. Ward

School of Engrg. Science, Plymouth Polytechnic, Plymouth PL4 8AA, UK, Shock Vib. Dig., 9 (8), pp 13-20 (Aug 1977) 3 figs, 42 refs

Key Words: Buildings, Seismic response, Reviews

This paper is concerned with structural dynamic problems involving buildings. Ground-borne disturbances including earthquakes, nuclear explosions, construction activities and vehicular traffic are discussed. Air-borne disturbances including wind and overpressures due to explosions are reviewed. Finally, thermal loads are included in the paper.

77-2059

Beam Vibrations - A Review

H. Wagner and V. Ramamurti

Indian Inst. of Tech., Madras, India, Shock Vib. Dig., 9 (9), pp 17-24 (Sept 1977) 115 refs

Key Words: Beams, Vibration response, Reviews

Most structural elements encountered in practice can be treated as beams sacrificing little accuracy. For this reason, this review article summarizes work on the vibration of beams since 1973.

77-2060

Turbomachinery Vibration

J.F. Traexler

Steam Turbine Div., Lester Branch, Westinghouse Electric Corp., Philadelphia, PA 19113, Shock Vib. Dig., 9 (8), pp 3-10 (Aug 1977) 8 figs

Key Words: Turbomachinery, Steam turbines, Vibration response, Rotors, Reviews

This article is concerned with turbomachinery vibrations, particularly those that occur in large steam turbines at central station power plants. Rotor dynamics and blading are reviewed.

77-2061

Exhaust Noise and Its Control - A Review

M.L. Munjal

Dept. of Mech. Engrg., Indian Inst. of Science, Bangalore - 12, India, Shock Vib. Dig., 9 (8), pp 21-32 (Aug 1977) 5 figs, 41 refs

Key Words: Mufflers, Noise reduction, Reviews

This article describes recent developments in the field of analysis and design of exhaust mufflers. The article is concerned only with exhaust noise.

77-2062

Acoustic Vibration of Structures in Liquids

D. Firth

Risley Engrg. and Materials Lab., United Kingdom Atomic Energy Authority, Risley, Warrington WA3 6AT, UK, Shock Vib. Dig., 9 (9), pp 3-7 (Sept 1977) 33 refs

Key Words: Submerged structures, Fluid-induced excitation, Acoustic excitation, Plates, Ducts, Reviews

This article outlines the physics of the vibration of an elastic structure excited by sound waves in a liquid in contact with the structure. The historical background is summarized, and some recent literature is described. Examples include plates, ducts, and complicated engineering systems. Possible future developments are suggested.

77-2063

Some Transient Problems of Structures Interacting with Fluid

D. Krajcinovic

Dept. of Materials Engrg., Univ. of Illinois at Chicago Circle, Chicago, IL, Shock Vib. Dig., 9 (9), pp 9-16 (Sept 1977) 5 figs, 29 refs

Key Words: Interaction: structure-fluid, Transient response, Reviews

This paper is a general review of transient interaction problems involving either a constant wetted surface or an expanding or receding wetted surface.

MODAL ANALYSIS AND SYNTHESIS

(See No. 2072)

COMPUTER PROGRAMS

GENERAL

77-2064

A FORTRAN IV Computer Program for the Time Domain Analysis of the Two-Dimensional Dynamic Motions of General Buoy-Cable-Body Systems

H.T. Wang

David W. Taylor Naval Ship Res. and Dev. Center, Bethesda, MD., Rept. No. DTNSRDC-77-0046, 95 pp (June 1977)
AD-A041 049/8GA

Key Words: Computer programs, Buoys, Cables, Dynamic response

The present report gives a detailed description of Program CABUOY, which analyzes in the time domain the two-dimensional dynamic behavior of general ocean cable systems consisting of a surface buoy, connecting cable, and intermediate bodies. The equations which model the motions of the surface waves and the various components of the cable system are presented, and the subroutines of the program are briefly outlined. Instructions on the use of the program include a listing of the input READ statements, definitions of the input variables, and a number of comments on the entering of input data. Several sample problems are given to illustrate use of the program, the output of the program, and computer costs for a range of cases. The listing of the program is given in the appendix.

77-2065

Computer Programs for the Calculation of Flexural Vibration of Turbomachinery Shafts (Programm-system zur Berechnung von Biegeschwingungszusständen an Turbomaschinenwellen)

E. Thomas and K.-H. Schubert

VEB Bergmann Borsig/Görlitzer Maschinenbau, West Berlin, German Democratic Republic, Maschinenbau-technik, 26 (7), pp 322-326 (July 1977) 6 figs, 6 refs

(In German)

Key Words: Computer programs, Turbomachinery, Shafts

The article describes computer programs for the calculation of vibration behavior of turbomachinery shafts, available at the VEB Bergmann Borsig/Görlitzer Maschinenbau. The aim of the calculations in recent years has been to achieve a high degree of automation of the turbomachinery shaft vibration calculation taking the actual conditions as much as possible into consideration.

77-2066

Nonlinear Analysis of Frame Structures Subjected to Blast Overpressures

W. Stea, G. Tseng, D. Kossover, S. Weissman, and N. Dobbs

Ammann and Whitney, New York, NY, Rept. No. ARLCD-CR-77008, 440 pp (May 1977)
AD-A040 708/0GA

Key Words: Computer programs, Frames, Buildings, Blast resistant structures

In modern day explosive manufacturing and LAP facilities, many of the structural steel buildings will be required to provide protection for personnel and/or equipment against the effects of HE-type explosions. Therefore, computer program entitled 'Dynamic Nonlinear Frame Analysis' (DYNFA) has been developed whereby the responses of frame structures subjected to blast loadings can be determined. This report contains the background for the development of DYNFA as well as the equations and procedures necessary for its use. The report also contains example problems illustrating the use of DYNFA for the design of blast-resistant frame structure.

77-2067

First Report on Capabilities of Dynamic Structural Analysis by the Strudl Program (Primo Rapporto Sulle Capacita Di Analisi Dinamica Dello Strudl)

B. Atzori and F. Fresa

Ist. di Construzione di Macchine, Bari Univ., Italy,
Rept. No. 76-2, 18 pp (Oct 1976)
(In Italian)
N77-26551

Key Words: Computer programs, Frames, Dynamic structural analysis

The capabilities of dynamic structural analysis by the STRUDL 2 program were studied. The case of frame analysis was examined for checking the validity of the results. Several factors, such as the influence of the number of elements on the approximation of the results and the CPU time necessary to solve some typical cases, were also investigated.

77-2068

A Sparsity-Oriented Approach to the Dynamic Analysis and Design of Mechanical Systems -- Part 1
N. Orlandea, M.A. Chace, and D.A. Calahan
Dept. of Mech. Engrg., Iowa State Univ., Ames, IA,
J. Engr. Indus., Trans. ASME, 99 (3), pp 773-779
(Aug 1977) 7 figs, 2 tables, 14 refs

Key Words: Computer programs, Computer-aided design, Suspension systems (vehicles), Landing gear

The work described herein is an extension of sparse matrix and stiff integrated numerical algorithms used for the simulation of electrical circuits and three-dimensional mechanical dynamic systems. By applying these algorithms big sets of sparse linear equations can be solved efficiently, and the numerical instability associated with widely split eigenvalues can be avoided. The new numerical methods affect even the initial formulation for these problems. In this paper, the equations of motion and constraints (Part 1) and the force function of springs and dampers (Part 2) are set up, and the numerical solutions for static, transient, and linearized types of analysis as well as the modal optimization algorithms are implemented in the ADAMS (automatic dynamic analysis of mechanical systems) computer program for simulation of three-dimensional mechanical systems (Part 2). The paper concludes with two examples: computer simulation of the front suspension of a 1973 Chevrolet Malibu and computer simulation of the landing gear of a Boeing 747 airplane. The efficiency of simulation and comparison with experimental results are given in tabular form.

77-2069

A Sparsity-Oriented Approach to the Dynamic Analysis and Design of Mechanical Systems -- Part 2
N. Orlandea, D.A. Calahan, and M.A. Chace
Dept. of Mech. Engrg., Iowa State Univ., Ames, IA,
J. Engr. Indus., Trans. ASME, 99 (3), pp 780-784

(Aug 1977) 3 figs, 2 tables, 9 refs

Key Words: Computer programs, Computer-aided design, Suspension systems (vehicles), Landing gear

The work described herein is an extension of sparse matrix and stiff integrated numerical algorithms used for the simulation of electrical circuits and three-dimensional mechanical dynamic systems. By applying these algorithms, big sets of sparse linear equations can be solved efficiently, and the numerical instability associated with widely split eigenvalues can be avoided. The new numerical methods affect even the initial formulation for these problems. In this paper, the equations of motion and constraints (Part 1) and the force function of springs and dampers (Part 2) are set up, and the numerical solutions for static, transient, and linearized types of analysis as well as the model optimization algorithms are implemented in the ADAMS (automatic dynamic analysis of mechanical systems) computer program for simulation of three-dimensional mechanical systems (Part 2). The paper concludes with two examples: computer simulation of the front suspension of a 1973 Chevrolet Malibu and computer simulation of the landing gear of a Boeing 747 airplane. The efficiency of simulation and comparison with experimental results are given in tabular form.

77-2070

Torsional Vibration Calculations of Machine Tool Drives (Berechnung des Torsionsschwingungsverhaltens von Werkzeugmaschinenantrieben)

R. Böhm
Konstruktion, 29 (7), pp 259-264 (July 1977)
13 figs, 4 refs
(In German)

Key Words: Computer programs, Torsional vibration, Machine tools, Gear drives

Gear drives - especially spur gear drives - are the most commonly used main drives in machine tools. Earlier investigations have shown that the main drive has a very strong effect on the stability of machine tool. In the article a computer program BEIGE for calculation of torsional frequency and the shape of vibration is described, which requires as input only data taken from construction drawings. Experimental data confirm a sufficient accuracy of the method.

77-2071

Modal Frequency and Random Response of the Airbus A300B Antenna
H. Goedel and F. Weiss

Messerschmitt-Boelkow-Blohm G.m.b.H., Ottobrunn, West Germany, Rept. No. UFE-1242-0, 14 pp (Apr 27, 1976) refs

N77-25378

Key Words: Antennas, Computer programs, Frequency response, Shells

Using the NASTRAN program system a computation of vibration and response was carried out for the ADF (Automatic Direction Finder) of the Airbus A300B in order to estimate the stress level within the scope of service life considerations. Using RIGID FORMAT 3 for normal mode computations and RIGID FORMAT 11 for power spectral density analysis, it was possible to achieve the actually obtained results for frequency responses in a simple way by means of the NASTRAN system.

77-2072

Stiffness Coupling Application to Modal Synthesis Program. Users Guide

E.J. Kuhar

General Electric Co., Philadelphia, PA., Rept. No. NASA-CR-145197, 26 pp (1976)
N77-25575

Key Words: Computer programs, Modal synthesis, Stiffness methods, Matrix methods

A FORTRAN IV computer program used to perform modal synthesis of structures by stiffness coupling, using the dynamic transformation method is described. The program was named SCAMP (Stiffness Coupling Approach Modal-Synthesis Program). The program begins with the entry of a substructure's physical mode shapes and eigenvalues or a substructure's mass and stiffness matrix. If the mass and stiffness matrices are entered, the eigen problem for the individual substructure is solved. Provisions are included for a maximum of 20 substructures which are coupled by stiffness matrix springs.

77-2073

A FORTRAN Program to Extract Static and Dynamic Moments from Free Oscillations in a Wind Tunnel

R.L. Pope

Weapons Research Establishment, Salisbury, Australia, Rept. No. WRE-TN-1729(WR/D), 42 pp (Dec 1976) refs
N77-25093

Key Words: Computer programs, Parameter identification, Wind tunnel tests

A FORTRAN program was developed using the parameter estimation technique to extract the static pitching moment

and the dynamic pitch damping moment from incidence measurements taken during planar oscillations of a model in a wind tunnel. The advantage of the parameter estimation method of analysis in this particular case is its ability to treat highly nonlinear forms of the static pitching moment. Comparisons are made with other wind tunnel measurements. A listing of the program and a sample run are included.

ENVIRONMENTS

ACOUSTIC

(Also see Nos. 2055, 2056, 2108, 2153, 2158, 2160)

77-2074

Acoustic Diffraction. Part 1. Plane Diffractors and Wedges

E.J. Skudrzyk, S.I. Hayek, and A.D. Stuart

Applied Research Lab., Pennsylvania State Univ., University Park, PA., Rept. No. TM-73-109-Pt-1, 160 pp (May 14, 1973)
AD-A040 668/6GA

Key Words: Acoustic diffraction

This memorandum documents the theoretical investigations in the Acoustic Diffraction Program. This report discusses the acoustic diffraction and backscattering phenomena for plane and wedge scatterers which are insonified by plane or point sources. The theories of diffraction used in this report are those of the approximate integral representations of Kirchoff-Rubinowicz. Those were compared with the geometrical theory of diffraction (GTD) which is developed by J.B. Keller, and is based on the ray theory.

77-2075

Noise Due to the Interaction of Boundary Layer Turbulence with a Marine Propulsor or an Aircraft Compressor

N. Moiseev, B. Lakshminarayana, and D.E. Thompson
Applied Research Lab., Pennsylvania State Univ., University Park, PA., Rept. No. TM-76-258, 122 pp (Oct 11, 1976)
AD-A040 946/6GA

Key Words: Noise generation, Rotor blades, Compressors, Propulsion systems

The sound generated by the interaction of inlet boundary layer turbulence with a rotating blade row is investigated. To experimentally study this radiated sound, an existing aeroacoustic facility was modified to produce the inflows desired. The rotor was operated in air with different blade space-to-chord ratios, different flow coefficients and different anisotropic, nonhomogeneous turbulent inflows. The inflows ingested are: natural boundary layer on hub and annulus wall, a tripped boundary layer on the hub, and a fully developed boundary layer on the hub. The turbulence intensities and length scales were altered by placing a grid at the inlet.

77-2076

Industrial Noise Control: Putting it all Together

T.D. Miller

Donley, Miller and Nowikas, Inc., 56 State Highway 10, East Hawover, NJ 07936, Noise Control Engr., 9 (1), pp 24-31 (July/Aug 1977) 7 figs, 1 table, 5 refs

Key Words: Noise control, Industrial facilities, Human response, Regulations

Industrial noise control has two fundamental objectives: to meet the requirements of federal law and to protect employees' hearing. The author outlines a total noise control program, and details some of the steps necessary to ensure that these goals are successfully met at minimum cost.

77-2077

Shielding Highway Noise

Z. Maekawa

Environmental Acoustics Lab., Kobe Univ., Rokko, Kobe, 657, Japan, Noise Control Engr., 9 (1), pp 38-44 (July/Aug 1977) 12 figs, 14 refs

Key Words: Noise barriers, Traffic noise

One of the most widespread problems in environmental acoustics is the control of road traffic noise. In urban areas and in the vicinity of residential districts especially, this has become an extremely serious issue. The author reviews typical methods of noise shielding, presents new results of experimental studies, and introduces some theoretical approaches.

77-2078

Two Experiments on the Perceived Noisiness of Periodically Intermittent Sounds

I. Pollack and R.M. Garrett

Dept. of Architecture, Muroran Inst. of Tech., 27 Mizumoto-cho, Muroran, Hokkaido, Japan 050, Noise Control Engr., 9 (1), pp 16-23 (July/Aug 1977) 10 figs, 4 tables, 15 refs

Key Words: Noise tolerance, Human response

The author describes a study aimed at clarifying the nature of the perceived noisiness of intermittent sounds, in order to establish an efficient method of assessment. Experimental results indicate that loudness and noisiness are different qualities. Further research delineates the structure of human response to these sounds.

PERIODIC

77-2079

A New Method for Predicting Response in Complex Linear Systems II

J.L. Bogdanoff, K. Kayser, and W. Krieger

School of Aeronautics and Astronautics, Purdue Univ., West Lafayette, IN 47907, J. Sound Vib., 53 (4), pp 459-469 (1977) 8 figs, 2 tables, 6 refs Sponsored by NASA, Marshall Space Flight Center

Key Words: Linear systems, Random excitation, Steady state excitation, Lumped parameter method

A new method is presented for response estimation in complex lumped parameter linear systems under random or deterministic steady state excitation. The essence of the method is the use of relaxation procedures with a suitable error function to find the estimated response; natural frequencies and normal modes are not computed. For a 45 degree of freedom system, and two relaxation procedures, convergence studies are made. Frequency response estimates are made.

RANDOM

(Also see No. 2079)

77-2080

A Probabilistic Model for a Randomly Excited Flow

Y.K. Gayed, M.R. Haddara, and A.H.A. Baghadi Dept. of Mech. Engrg., Cairo Univ., Cairo, Egypt, Appl. Math. Modeling, 1 (6), pp 299-309 (Sept 1977) 9 figs, 1 table, 11 refs

Key Words: Hydroelectric power plants, Transient response, Random response, Mathematical models, Probability theory

This work concerns a probabilistic model of the random problem, whose solution gives the distribution and probability density functions of the variables involved, namely the pressures, velocities and surge tank oscillation. Order statistical methods were also used to estimate the probability of occurrence of extreme head fluctuations.

SEISMIC

(Also see Nos. 2097, 2151, 2169, 2170, 2171, 2185)

77-2081

Learning from Earthquakes. 1977 Planning and Field Guides

Earthquake Engrg. Research Inst., California Univ., Los Angeles, CA., Rept. No. NSF/RA-770081, 221 pp (1977)
PB-268 083/3GA

Key Words: Earthquake damage

The aim is to maximize the learning to be gained from investigations following future destructive earthquakes. The Guides are meant for use in the planning and field execution of such investigations. Through their use, both the afflicted communities and the investigators can understand how to participate in the investigation and what information is of greatest value.

77-2082

The Earthquake Response of Deteriorating Systems

N.C. Gates
Ph.D. Thesis, California Inst. of Tech., 140 pp (1977)
UM 77-19,980

Key Words: Linear systems, Earthquake response, Approximation methods, Stiffness methods, Energy methods

This thesis is concerned with the earthquake response of deteriorating systems. A model for stiffness degrading or deteriorating systems is used to describe six different single-degree-of-freedom systems. A numerical investigation of the response of these six systems is performed using an ensemble of twelve earthquakes. The response is studied at nine nominal periods of oscillation. The numerical results are presented as response spectra corresponding to six different ductilities. An approximate analytical method for calculating the earthquake response of deteriorating systems from a linear response spectrum is presented. The method, called the average stiffness and energy method, is based upon the premise that a linear system may be defined which is in some sense equivalent to the deteriorating system. The criterion for equivalence in this method is that the average stiffness of the deteriorating system be equal to the stiffness

of the linear system and the average energy dissipated by the linear system be the same as the average energy dissipated by the deteriorating system. The new analytical method is compared to existing methods. Comparison with the numerical results is also made. Based upon these comparisons, it is concluded that the average stiffness and energy method represents a significant improvement over currently available methods for predicting the earthquake response of deteriorating and nondeteriorating systems.

77-2083

Investigation of the Inelastic Characteristics of a Steel Frame Using System Identification and Shaking Table Experiments

V.C. Matzen
Ph.D. Thesis, Univ. of California, Berkeley, 127 pp (1976)
UM 77-15,782

Key Words: Framed structures, Seismic response, System identification, Experimental results

In this dissertation, system identification is used to formulate a realistic nonlinear mathematical model to represent the seismic behavior of a single story steel structure. With this model and the parameters established for it, the energy absorbing characteristics of the structure are investigated. During this study, system identification itself is examined to determine how it can be better utilized in structural engineering. There are three major parts to this research. The first is the mathematical development of system identification to meet the particular needs of this problem. The second part of the research involved shaking table experiments in which a single story steel frame was subjected to several earthquake excitations. The last part of the research is the use of test data in the identification program to establish the four parameters in the mathematical model. When different values are used for T, parameter sets are established which give the best model response for that amount of test data. The resulting sets of parameters reflect the way in which the properties of the structure change during the excitation.

77-2084

Performance and Analysis of Earth Dams During Strong Earthquakes

F.I. Makdisi
Ph.D. Thesis, Univ. of California, Berkeley, 248 pp (1976)
UM 77-15,778

Key Words: Dams, Earthquake response

An investigation into the behavior of a number of earth

dams that were severely shaken during the San Francisco 1906 earthquake was undertaken to identify the factors contributing to their adequate performance. It was found that the majority of these embankments consisted of predominantly clay soils. On the basis of the knowledge of the behavior of clays under cyclic loading conditions, it is shown that the clayey nature of these embankments was the significant factor contributing to their stability during the earthquake. In addition, the contrasting behavior of sandy embankments is demonstrated by studying the failure and near failure of a number of embankments during four other earthquakes in California and Japan.

SHOCK

(Also see Nos. 2048, 2066, 2150, 2186, 2188, 2189)

77-2085

Surface Waves Generated by Shallow Underwater Explosions

A. Falade

Ph.D. Thesis, Univ. of California, Berkeley, 93 pp (1976)

UM 77-15,679

Key Words: Underwater explosions, Explosion effects

In this report, surface water waves generated by surface and near surface point explosions are calculated. Taking impulse distribution imparted at the water surface by the explosion as the overriding mechanism for transferring energy of the explosive to surface wave motion, the linearized theory of Kranzer and Keller is used to obtain the wave displacement in the far field.

GENERAL WEAPON

77-2086

Parametric Resonance in Gun Tubes

T.E. Simkins

Watervliet Arsenal, NY, Rept. No. WVT-TR-77009, 70 pp (Feb 1977)
AD-A040 677/7GA

Key Words: Gun barrels, Parametric resonance

This work examines the likelihood of encountering parametric resonance in gun tubes. The resonance is induced conceptually by the periodic changes in transverse stiffness induced by the axial vibrations resulting from a single application of ballistic pressure - 'single round parametric resonance', the periodic applications of ballistic pressure such as

encountered in an automatic weapon - 'multiple round parametric resonance'.

TRANSPORTATION

(Also see Nos. 2186, 2188, 2189)

77-2087

Experimental Designs and Psychometric Techniques for the Study of Ride Quality

M.D. Havron and R.A. Westin

ENSCO, Inc., Springfield, VA., Rept. No. DOT-TSC-OST-76-54, 301 pp (May 1977)
PB-268 584/0GA

Key Words: Transportation vehicles, Ride dynamics, Human response, Statistical analysis

A major variable in both the cost of any new transportation system and rider acceptance of the system is the ride quality of its vehicles. At this time, there exists no set of objective criteria which would allow the transportation system designer to determine what level of ride quality would be considered acceptable by a wide variety of potential passengers. The purpose of the study was to establish statistically acceptable techniques for the development of methods for relating physical measures of vehicle vibration to passenger estimates of ride quality.

PHENOMENOLOGY

DAMPING

(Also see Nos. 2107, 2192)

77-2088

Tuned Mass Dampers for Buildings

R.J. McNamara

Gillum-Colaco Consulting Struct. Engrs., Cambridge, MA., ASCE J. Struc. Div., 103⁺ (ST9), pp 1785-1798 (Sept 1977) 13 figs, 14 refs

Key Words: Tuned dampers, Buildings, Single degree of freedom systems

Tuned mass dampers attached to single degree-of-freedom systems representing tall buildings are studied. System equations are formulated and solved for various input forcing

functions. Design parameters of the damper are varied to study the response reduction. Experimental wind tunnel results are presented, and a practical application of a large-scale damper is illustrated.

77-2089

The Damping of Structural Vibration by Rotational Slip in Joints

C.F. Beards and J.L. Williams

Dept. of Mech. Engrg., Imperial College of Science and Tech., London SW7 2BX, UK, J. Sound Vib., 53 (3), pp 333-340 (Aug 8, 1977) 5 figs, 3 tables, 9 refs

Key Words: Slip joints, Coulomb friction, Computer programs

Interfacial slip in joints is the major contributor to the inherent damping of most fabricated structures. By fastening joints tightly enough to prohibit translational slip, but not tightly enough to prohibit rotational slip (thereby making only a small sacrifice in static stiffness), it is shown, both experimentally and theoretically, that a useful increase in the inherent damping in a structure can be achieved, provided an optimum joint load is maintained. The analysis is simplified by using a general dynamic analysis computer program with a sub-program to model the friction joint.

77-2090

Some Comments on the Estimation of Resonant Peak Amplitudes

R.E.D. Bishop

Dept. of Mech. Engrg., Univ. College London, Torrington Place, London WC1E 7JE, UK, Ing. Arch., 45 (5/6), pp 331-336 (1976) 4 figs, 5 refs

Key Words: Resonant response, Damped structures, Forced vibration

In a recently published paper, a way of estimating resonant responses of a damped system by means of calculations for the undamped system was suggested. No reference was made to the existing literature on the theory of forced vibration. The object of the present paper is to show how his approach fits in and, in particular, to illustrate what it implies in terms of polar response plots.

77-2091

Subsynchronous Resonance in Power Systems: Damping of Torsional Oscillations

K.T. Khu

Ph.D. Thesis, Iowa State Univ., 154 pp (1977) UM 77-16,962

Key Words: Electric generators, Vibration resonance, Torsional vibrations, Self-excited vibrations, Hunting, Vibration damping

Studies of subsynchronous resonance phenomena are conducted in a power system composed of a tandem-compound steam turbo-generator set connected to an infinite bus via a series capacitor compensated transmission line. Complete detailed representation of the electromechanical system has confirmed the existence of (n-1) modes of oscillation, where n is the number of lumped masses of the shaft, as well as the existence of super- and subsynchronous components in the electrical network. The eigenvalue method of analysis is used to study the interaction between the mechanical and electrical networks under small perturbations, and to identify the conditions in which the system would be subjected to torsional interaction, self-excitation, and hunting. Transient analysis is carried out on an analog computer to observe the electrical quantities and the torques of the various sections of the shaft before, during, and after a three-phase fault is applied.

ELASTIC

77-2092

Dynamic Stresses Produced in an Elastic Half Space by Reciprocally Moving Surface Loads

T. Ohyoshi

Mining College, Akita Univ., Akita, Japan, Bull. JSME, 20 (145), pp 777-784 (July 1977) 10 figs, 5 refs

Key Words: Elastic properties, Half space, Moving loads

In studies of moving load problems, Galilean or Laplacian transformations have been commonly used by several previous investigators to construct the solutions. In this paper analytical techniques of superposition of harmonic vibrations are available because the elements composing an elastic half space are excited periodically by reciprocating surface loads.

77-2093

Elastodynamic Analysis of a Completely Elastic System

D. Kohli, D. Hunter, and G.N. Sandor

Univ. of Wisconsin, Milwaukee, WI., J. Engr. Indus., Trans. ASME, 99 (3), pp 604-609 (Aug 1977) 3 figs, 1 table, 16 refs

Key Words: Slider crank mechanisms, Elastodynamic response, Transverse shear deformation effects, Rotatory inertia effects

The completely elastic system considered for this vibration analysis consists of an offset slider-crank mechanism having elastic supports and mountings of the mechanism permitting translational vibrations of the shafts and supports, elastic shafts permitting torsional vibrations, elastic links of the mechanism which deform due to external or internal body forces and allow flexural and axial vibrations. Both the effect of the deformations caused by the inertia forces in the mechanism links, shafts, and supports and the effect of change in the inertia forces due to these deformations are taken into account in constructing a general mathematical model for conducting elastodynamic analysis.

FLUID

(See No. 2062)

SOIL

(Also see No. 2185)

77-2094

Dynamic Torsional Response of Foundations on Layered Media

A. Prodanovic

Ph.D. Thesis, Rice Univ., 278 pp (1977)
UM 77-19,285

Key Words: Footings, Foundations, Torsional response, Layered materials

A study is made of the steady-state harmonic torsional response of a rigid circular footing perfectly bonded to the surface of a layered elastic or viscoelastic medium, the footing being excited either kinematically or under the action of a torque. The supporting medium is assumed to consist of a finite number of horizontal layers of constant thickness overlying a homogeneous half-space. Primary attention is given to the problems involving a single layer over a homogeneous half-space and a stratum over a rigid base; the homogeneous half-space is also considered as a limiting case.

77-2095

Dynamics of Certain Structure-Foundation Interacting Systems

J.B. Valdivieso

Ph.D. Thesis, Rice Univ., 227 pp (1977)
UM 77-19,297

Key Words: Interaction: structure-foundation

An analytical investigation of three interrelated problems in the general area of structure-foundation interaction is conducted. The effects of the presence of a substantial foundation mass on the response of interacting systems is initially studied. The foundation medium is assumed to be a halfspace with elastic or viscoelastic properties. Attention is given to the effects of foundation mass on the magnitude of the forces developed during motion since these generally govern the structural design. The applicability of the use of a Single Degree of Freedom equivalent oscillator to predict the dynamic behavior of a soil-structure interacting system with a finite foundation mass is assessed.

77-2096

Dynamic Response of Friction Piles

C.-S. Chon

Ph.D. Thesis, The Univ. of Michigan, 232 pp (1977)
UM 77-17,968

Key Words: Interaction: soil-structure, Pile structures

The influence of several "soil-pile interaction" parameters on the dynamic and static response of single friction piles to lateral loads were studied by performing model pile tests and comparing the results with theoretical analyses. Both dynamic and static model pile tests were performed in a specially constructed facility which was designed to operate as a "quicksand" tank. The quicksand operation provided for rapid and easy reconstitution of the fine, uniform sand to preselected conditions before each test.

77-2097

Seismic Response of Axisymmetric Soil-Structure Systems

E. Berger

Ph.D. Thesis, Univ. of California, Berkeley, 189 pp (1976)
UM 77-15,607

Key Words: Interaction: soil-structure, Seismic response, Finite element techniques, Computer programs, Nuclear power plants

The accuracy of seismic response computations made with two-dimensional finite element methods of analysis applied to three-dimensional soil-structure systems is investigated. The three-dimensional soil-structure system is modeled by an axisymmetric finite element model while the equivalent two-dimensional system is represented by a plane strain model. A finite element computer code ALUSH is developed which computes the seismic response of axisymmetric soil-structure systems subjected to horizontal, vertical and

rotational earthquake input motions. The nonlinear stress-strain behavior of soil masses subjected to strong earthquake motions and the frequency independent nature of the damping characteristics of soils are considered in the method of analysis by use of equivalent linear method and the complex response method, respectively.

77-2098

Unified Boundary for Finite Dynamic Models

W. White, S. Valliappan, and I.K. Lee

Dept. of Civil Engrg. Materials, The Univ. of New South Wales, Kensington, New South Wales, Australia, ASCE J. Engr. Mech. Div., 103 (EM5), pp 949-964 (Oct 1977) 4 figs, 3 tables, 8 refs

Key Words: Soils, Dynamic response, Finite element technique, Energy absorption

The finite element analysis of dynamic problems in an infinite, isotropic medium is examined. To simulate the physically infinite system by a finite model, an energy absorbing boundary is proposed. This boundary is frequency independent and proves to be very efficient in absorbing stress waves. The boundary constants are calculated for the particular cases of plane strain and axisymmetry for isotropic materials.

77-2099

Hydrodynamic Pressure in Semicylindrical Reservoir

F.J. Sanchez-Sesma and E. Rosenblueth

Instituto de Ingeniería, Universidad Nacional Autónoma de México, México, ASCE J. Engr. Mech. Div., 103 (EM5), pp 913-919 (Oct 1977) 4 figs, 3 tables, 11 refs

Key Words: Dams, Modal analysis, Seismic design, Hydrodynamic excitation

Solutions are presented for modal analysis of hydrodynamic pressures generated by the three translational seismic components - longitudinal, vertical, and transverse - on a dam limiting a semicircular cylindrical reservoir. The main purpose is to show the influence of the cross-sectional shape of the reservoir in the hydrodynamic responses. Results are compared with those for rectangular cross section.

VISCOELASTIC

77-2100

Design of a Viscoelastic Dynamic Absorber for Machine Tool Applications

G.L. Nessler, D.L. Brown, D.C. Stouffer, and K.C. Maddox

Appl. Dynamics & Acoustics Section, Battelle Columbus Labs., Columbus, OH, J. Engr. Indus., Trans. ASME, 99 (3), pp 620-623 (Aug 1977) 5 figs, 11 refs

Key Words: Machine tools, Viscoelastic damping

The design equations are developed for a viscoelastic dynamic absorber in uniaxial compression. The dependence of mechanical properties of the absorber on frequency, temperature, and preload are developed through an extension of the thermorheologically simple theory of linear viscoelasticity. An approximation of the exact boundary value problem is made in order to develop practical design criteria for the size and shape of the absorber element. The results of the experimental program for the constitutive equation are included. A dynamic absorber is designed to control a self-excited lathe chatter problem and a significant improvement is demonstrated.

EXPERIMENTATION

DIAGNOSTICS

(Also see No. 2126)

77-2101

Increase Plant Availability with Trend Monitoring

E.G. Filetti and P.R. Trumpler

Energy Technology, Inc., West Chester, PA, Hydrocarbon Processing, 56 (9), pp 233-240 (Sept 1977) 5 figs, 2 refs

Key Words: Diagnostic techniques, Machinery vibration, Critical speed, Whirling

Trend monitoring is a modern engineering method designed to minimize unscheduled process plant shutdowns by anticipating malfunctions in on-line machines. The onset of machine problems is usually detected as an increase in vibration level. Two particularly important machine characteristics, lateral critical speeds and whirl, are discussed in some detail. Several applications are also described.

77-2102

A Survey of Design Methods for Failure Detection in Dynamic Systems

A.S. Wilksky

Electronic Systems Lab., Massachusetts Inst. of Tech., Cambridge, MA., In: AGARD Integrity in Electron. Flight Control Systems, 14 pp (Apr 1977) refs (N77-25055)
N77-25060

Key Words: Diagnostic techniques, Dynamic systems, Nonlinear systems

A number of methods for the detection of abrupt changes (such as failures) are surveyed in stochastic dynamical systems. The class of linear systems is concentrated, but the basic concepts, if not the detailed analyses, carry over to other elements of systems. The methods range from the design of specific failure-sensitive filters, to the use of statistical tests on filter innovations, to the development of jump process formulations. Tradeoffs in complexity versus performance are discussed.

77-2103

What Can Mini-Computers do for Machinery Reliability?

R.G. Harker

Bently Nevada Corp., Minden, NV, Hydrocarbon Processing, 56 (8), pp 137-143 (Aug 1977) 11 figs

Key Words: Diagnostic techniques

As major turbomachinery trains become more complex and critical, condition monitoring for maximum reliability becomes more important. Dedicated mini-computer systems appear to be the coming way to perform this task.

FACILITIES

77-2104

An Historical View of Dynamic Testing

H.C. Pusey

Naval Research Lab., Shock and Vibration Information Center, Washington, D.C., J. Environ. Sci., 20 (5), pp 9-14 (Sept/Oct 1977) 83 refs

Key Words: Dynamic testing, Reviews

Developments in the field of dynamic testing over the past thirty years are examined. Assessment of present capabilities and future needs leads to the conclusion that the problems to be solved are more managerial than technical. Some controversial questions are posed with respect to dynamic tests and specifications.

INSTRUMENTATION

77-2105

New Electromagnetic Transducers for Recording Translations and Vibrations

B.Z. Kaplan

Dept. of Electrical Engrg., Ben Gurion Univ. of the Negev, Be'er Sheva, Israel, Israel J. Tech., 14 (4/5), pp 187-195 (1976) 10 figs, 12 refs

Key Words: Transducers, Measuring instruments, Recording instruments, Vibration measurement, Vibration recording

The paper discusses new instrumentation developed for measuring translations and vibrations of mechanical parts. Electromagnetic fields are employed for these measurements, and mechanical loading is, therefore, avoided. At first one-sided capacitive transducers are treated. Secondly, differential capacitive transducers are discussed. An electronic method is investigated by which the operation of such differential bridges can be maintained linear even if the deviation of the moving member from its central position was large. The last parts of the paper deal with microwave interferometric bridges. It is shown that movements of remote objects with amplitudes in the micrometer region can be recorded from distances of several meters.

TECHNIQUES

(Also see Nos. 2087, 2159)

77-2106

Application of Modal Testing Techniques to Solve Vibration Problems in Machinery Supporting Structures

J.W. Martz and T. Leist

Structural Dynamics Research Corp., Cincinnati, OH, ASME Paper No. 77-DE-16

Key Words: Testing techniques, Modal testing, Machine foundations

This paper describes the use of state-of-the-art testing techniques to solve vibration problems that result from design incompatibility between machinery and the machinery supporting structures. The general techniques of "mechanical impedance," or "modal" testing described herein have become widely used in the laboratory over the past several years to solve vibration problems in machine tools, automotive vehicles, construction, and agricultural machinery.

77-2107

A Forced-Vibration Technique for Measurement of Material Damping

R.F. Gibson and R. Plunkett

Dept. of Engrg. Science and Mech., and Engrg. Res. Inst., Iowa State Univ., Ames, IA 50011, Exptl. Mech., 17 (8), pp 297-302 (Aug 1977) 9 figs, 18 refs

Key Words: Measurement techniques, Material damping

This article describes a technique for measuring material damping in specimens under forced flexural vibration. Although the method was developed for testing fiber-reinforced composite materials, it could be used for any structural material. The test specimen is a double-cantilever beam clamped at its midpoint and excited in resonant flexural vibration by an electromagnetic shaker. Under steady state conditions, material damping is defined in terms of the ratio of input energy to strain energy stored in the specimen. If external losses are negligible, the input energy must equal the energy dissipated in the specimen. Input energy and strain energy are found from measured specimen dimensions, resonant frequency, input acceleration and bending strain. Problems associated with minimization of external energy losses in the apparatus and verification of measurements are discussed in detail. Measured damping of aluminum-alloy calibration specimens shows good agreement with calculated thermoelastic damping. Examples of measured damping showing amplitude and frequency dependence in fiber-reinforced plastic materials are presented.

77-2108

Characteristics and Calibration of Reference Sound Sources

P. Francois

Electricité de France, 1 Avenue General de Gaulle, 92141 Clamart, France, Noise Control Engr., 9 (1), pp 6-15 (July/Aug 1977) 9 figs, 3 tables, 8 refs

Key Words: Noise measurement, Measurement techniques

The reference sound source - a source of known acoustic power output - was developed in the United States in the mid-1950s. Several new devices to simplify the determination of sound power have emerged since 1970, and standards for the characteristics, calibration, and usage of these instruments are now being developed. Current sources and some proposed techniques for calibration are discussed by the author.

COMPONENTS

ABSORBERS

(See No. 2100)

BEAMS, STRINGS, RODS, BARS

(Also see Nos. 2059, 2064, 2065)

77-2109

The General Solution to the Classical Problem of Finite Euler Bernoulli Beam

M.Y. Hussaini and C.L. Amba-Rao

Ames Research Center, NASA, Moffett Field, CA., Rept. No. NASA-TM-X-73253; A-7076, 13 pp (June 1977)
N77-26533

Key Words: Beams, Bernoulli theory, Free vibration, Forced vibration, Winkler foundations, Viscous damping

An analytical solution is obtained for the problem of free and forced vibrations of a finite Euler Bernoulli beam with arbitrary (partially fixed) boundary conditions. The effects of linear viscous damping, Winkler foundation, constant axial tension, a concentrated mass, and an arbitrary forcing function are included in the analysis. No restriction is placed on the values of the parameters involved, and the solution presented here contains all cited previous solutions as special cases.

77-2110

Thin-Walled Curved Beam Finite Element

S.K. Chaudhuri and S. Shore

ASCE J. Engr. Mech. Div., 103 (EM5), pp 921-937 (Oct 1977) 9 figs, 5 tables, 21 refs

Key Words: Curved beams, Bridges, Moving loads

The generalized displacements and forces at the two nodes of the beam elements are: three translations and their corresponding forces, three rotations and their corresponding moments, the out-of-plane warping of the end cross section and its corresponding bi-moment. The solutions to the homogeneous differential equations governing the static deformation of curved beams along with kinematical boundary conditions are given. The stiffness matrix is formed by evaluating the stress resultants at the two ends of the element

corresponding to each unit generalized displacement. The method using the principle of virtual work to obtain the equivalent nodal forces due to external loading and the consistent mass matrix is outlined. Several examples are presented and comparisons made to demonstrate the accuracy and the usefulness of the element. This element has been successfully used in the finite element discretization of curved girders of horizontally curved highway bridges in studying the response of the bridges subjected to moving loads.

77-2111

Response of Beam to Stochastic Boundary Excitation

S.F. Masri and A. Aryafar

Dept. of Civil Engrg., Univ. of Southern California, Los Angeles, CA., ASCE J. Engr. Mech. Div., 103 (EM5), pp 807-822 (Oct 1977) 14 figs, 7 refs

Key Words: Beams, Boundary condition effects, Bernoulli-Euler method, Stochastic processes

A closed-form solution is presented for the covariance kernel of the transient response of a damped Bernoulli-Euler beam with arbitrary boundary conditions to correlated stochastic excitation applied at the boundaries. The analytical results are applied to the case where the autocorrelation function of the excitation resembles that of a wide class of input functions including earthquake excitations. The mean-square transient response at arbitrary locations along the beam is evaluated, and the effects of various system parameters are determined.

77-2112

Dynamic Responses of Viscoelastic Continuous Beams on Elastic Supports

K. Nagaya and Y. Hirano

Faculty of Engrg., Yamagata Univ., Yonezawa, Japan, Bull. JSME, 20 (145), pp 785-792 (July 1977) 10 figs, 12 refs

Key Words: Continuous beams, Viscoelastic properties, Elastic foundations

This paper deals with the vibration and the transient response problems of a viscoelastic continuous beam on non-periodic elastic supports. In the analysis, the restoring forces of the elastic supports are regarded as unknown external forces applied to the beam. The solution for the viscoelastic beam is obtained from the correspondence principle by applying the Laplace transform to the constitutive equation and the equation of motion for the elastic beam in terms of these unknown forces.

77-2113

Experimental Assessment of the Mindlin-McNiven Rod Theory

H.D. McNiven and Y. Mengi

Univ. of California, Berkeley, CA 94720, J. Acoust. Soc. Amer., 62 (3), pp 589-594 (Sept 1977) 8 figs, 6 refs

Key Words: Rods, Axisymmetric vibrations, Approximation methods, Experimental data

The three-mode theory due to Mindlin and McNiven, [J. Appl. Mech. 27, 145-151 (1960)] governing axisymmetric motions in a circular rod, is appraised by comparing responses predicted by it with experimental data obtained by Miklowitz and Nisewanger [J. Appl. Mech. 24, 240-244 (1957)]. The problem studied involves a semi-infinite rod, made of 24S-T aluminum alloy, subjected to pressure applied to the end of the rod. The two sets of responses are compared at various stations along the rod. To make the comparisons meaningful, it was necessary to recognize that the pressure applied experimentally had a finite rise time, however short; to make an estimate from the responses of what that rise time might be; and then apply this time distribution of pressure in evaluating the theoretical responses.

BEARINGS

77-2114

Consideration of the Negative Pressure Field at the Computation of Dynamic Loaded Radial Sliding Bearings. Model of a Fluid-Gas-Mixture in the Lubrication Gap (Einbeziehung des Unterdruckgebietes in die Berechnung dynamisch belasteter Radialgleitlager. Modell eines Flüssigkeits-Gas-Gemisches im Schmierspalt)

R. Wegmann

Wilhelm-Pieck-Universität Rostock, German Dem. Republic, Maschinenbautechnik, 26 (7), pp 320-321 (July 1977) 2 figs, 8 refs
(In German)

Key Words: Slider bearings, Dynamic response

The article shows that for the calculation of dynamically loaded sliding bearings it is necessary to consider the negative pressure field. The behavior of lubricants at low pressures is described and a model for the fluid-gas-mixture is set up.

77-2115

A Cantilever Mounted Resilient Pad Gas Bearing

I. Etsion

Lewis Res. Center, NASA, Cleveland, OH, Rept. No. NASA-CASE-LEW-12569-1, 12 pp (Apr 28, 1977) PAT-APPL-792 069/GA

Key Words: Gas bearings

The patent application relates to a gas-lubricated bearing employing at least one pad mounted on a rectangular cantilever beam to produce a lubricating wedge between the face of the pad and a moving surface. The load-carrying and stiffness characteristics of the pad are related to the dimensions and modulus of elasticity of the beam. The invention is applicable to a wide variety of types of hydrodynamic bearings.

BLADES

(Also see No. 2174)

77-2116

Wind Tunnel Tests of a Two Bladed Model Rotor to Evaluate the TAMI System in Descending Forward Flight

R.P. White, Jr.

Rasa Div., Systems Research Labs., Inc., Newport News, VA., Rept. No. NASA-CR-145195, 53 pp (May 1977) refs

N77-25080

Key Words: Rotor blades, Noise reduction, Vortex induced excitation

A research investigation was conducted to assess the potential of the Tip Air Mass Injection system in reducing the noise output during blade vortex interaction in descending low speed flight. In general it was concluded that the noise output due to blade vortex interaction can be reduced by 4 to 6 db with an equivalent power expenditure of approximately 14 percent of installed power.

77-2117

Flap/Lag Torsion Dynamics of a Uniform, Cantilever Rotor Blade in Hover

W. Johnson

Ames Res. Center, NASA, Moffett Field, CA., Rept. No. NASA-TM-73248, A-7063, 19 pp (May 1977) Sponsored in part by the U.S. Army Air Mobility Res. and Dev. Lab., Moffett Field, CA N77-26068

Key Words: Rotor blades, Dynamic stability

The dynamic stability of the flap/lag/torsion motion of a uniform, cantilever rotor blade in hover is calculated. The influence of blade collective pitch, lag frequency, torsional flexibility, structural coupling, and precone angle on the stability is examined. Good agreement is found with the results of an independent analytical investigation.

77-2118

Unsteady Hovering Wake Parameters Identified from Dynamic Model Tests. Part 1

K.H. Hohenemser and S.T. Crews

Dept. of Mech. Engrg., Washington Univ., St. Louis, MO., Rept. No. NASA-CR-152022, 120 pp (June 1977)

N77-26077

Key Words: Rotor blades, Parameter identification, Perturbation theory

The development of a 4-bladed model rotor is reported that can be excited with a simple eccentric mechanism in progressing and regressing modes with either harmonic or transient inputs. Parameter identification methods were applied to the problem of extracting parameters for linear perturbation models, including rotor dynamic inflow effects, from the measured blade flapping responses to transient pitch stirring excitations. These perturbation models were then used to predict blade flapping response to other pitch stirring transient inputs, and rotor wake and blade flapping responses to harmonic inputs. The viability and utility of using parameter identification methods for extracting the perturbation models from transients are demonstrated through these combined analytical and experimental studies.

DUCTS

(Also see No. 2062)

77-2119

Transmission of Sound Through Nonuniform Circular Ducts with Compressible Mean Flows

A.H. Nayfeh, B.S. Shaker, and J.E. Kaiser

Dept. of Engrg. Science and Mech., Virginia Polytechnic Inst. and State Univ., Blacksburg, VA., Rept. No. NASA-CR-145126, 66 pp (May 1977)

N77-25914

Key Words: Ducts, Sound transmission, Sound attenuation, Computer programs

An acoustic theory is developed to determine the sound transmission and attenuation through an infinite, hard-walled or lined, circular duct carrying compressible, sheared, mean flows and having a variable cross section. The theory is applicable to large as well as small axial variations, as long as the mean flow does not separate.

77-2120

Sound Attenuation in Multiply Lined Rectangular Ducts Including Effects of the Wall Impedance Discontinuities. Part 2: Liners in Parallel

W. Koch

Deutsche Forschungs- und Versuchsanstalt für Luft- und Raumfahrt, Goettingen, West Germany, Rept. No. ESA-TT-399, DLR-FB-76-58, 42 pp (Nov 11, 1976) refs
(In German)
N77-25917

Key Words: Ducts, Acoustic liners, Noise reduction

The problem of sound attenuation by a combination of two acoustic liners of finite length and of different wall impedance on opposite walls in an infinitely long rectangular duct was formulated as a Wiener-Hopf problem for zero mean flow. A coupled system of two generalized Wiener-Hopf equations was obtained and solved. Numerical results are given for a realistic wall impedance model. The influence of several liner parameters on sound attenuation is displayed graphically.

FRAMES, ARCHES

77-2121

The Steady State Response of Geometrically Non-Linear Shallow Arches

D. Hitchings and P. Ward

Dept. of Aeronautics, Imperial College of Science and Tech., London, UK, Intl. J. Numer. Methods Engr., 11 (8), pp 1261-1269 (1977) 5 figs, 9 refs

Key Words: Arches, Periodic response, Finite element technique

The non-linear steady state response of structures with curvature is investigated through the expository example of a shallow circular arch. A consistent mass finite element formulation is employed to derive the governing non-linear differential equations. These equations are solved by assuming a single mode expansion reducing the governing equations to the single degree-of-freedom Duffing's equation with a

quadratic term. The non-symmetric amplitude-frequency curve is derived and compared with results previously obtained by direct integration of the equations of motion.

GEARS

77-2122

Digital Simulation of Impact Phenomenon in Spur Gear Systems

R.C. Azar and F.R.E. Crossley

Dept. of Mech. Engrg., Western New England College, Springfield, MA., J. Engr. Indus., Trans. ASME, 99 (3), pp 792-798 (Aug 1977) 11 figs, 17 refs

Key Words: Gears, Shafts, Impact pairs, Digital simulation, Torsional vibrations

A digital simulation model is developed to represent a lightly geared torsional system consisting of a drive unit, spur gear pair and load connected by flexible shafts. A clearance model called an Impact Pair is used to represent the gear pair and includes the effects of backlash, time-varying stiffness and damping of the gear teeth and tooth-form error. Experimentally determined frequency spectra of the torsional oscillations of a gear-driven shaft have been plotted and reported on earlier. Similar frequency plots are obtained from the simulation study, and data from these plots are compared with the experimental results for a variety of parameter changes including shaft speed, backlash and load. Results indicate that the simulation model portrays reasonably well the torsional behavior of the output shaft.

77-2123

Dynamic Stability of a Two-Stage Gear Train Under the Influence of Variable Meshing Stiffnesses

G.V. Tordion and R. Gauvin

Dept. of Mech. Engrg., Laval Univ., Quebec, P.Q., Canada, J. Engr. Indus., Trans. ASME, 99 (3), pp 785-791 (Aug 1977) 11 figs, 4 refs

Key Words: Gears, Parametric excitation, Dynamic stability

In a two-stage gear train, the two meshing stiffnesses acting on the intermediate shaft produce parametric vibrations. Equations to find the principal and secondary regions of instability are given. Results showing the influence of the phase angle between both meshing stiffnesses are presented. An easy way to determine whether a certain operating condition lies in a stability or instability region is also suggested.

77-2124**Measurement and Evaluation of Geared Engine Noises (Messung und Beurteilung der Geräusche von Getriebemotoren)**

H. Greiner

Industrie-Anz., 98 (72), pp 1281-1284 (1976)
(In German)**Key Words:** Gears, Engine noise

The article describes the causes of gear noises in engines. Gear sizes, speed reduction, skew angle, profile offset, profile correction, concentricity of the shaft end, gear material and hardness, lubrication, relative loading are considered. The article also describes how the measured noises are evaluated, analysis and evaluation of noise frequencies, ISO noise rating, determining factors for noise intensity of geared motors, and decline of noise level with distance.

77-2125**Statistical Analysis of Dynamic Loads on Spur Gear Teeth**

T. Tobe and K. Sato

Faculty of Engrg., Tohoku Univ., Sendai, Japan,
Bull. JSME, 20 (145), pp 882-888 (July 1977)
15 figs, 11 refs**Key Words:** Gears, Dynamic loads, Statistical analysis

Analysis of transmission error curve of a pair of gears measured by a single flank meshing tester shows that the error can be separated into harmonic components and random ones. In this paper the effect of the random components of the error on dynamic loads is analyzed theoretically. One example of numerical result is shown.

LINKAGES

(Also see No. 2089)

77-2126**The Theory of Torque, Shaking Force, and Shaking Moment Balancing of Four Link Mechanisms**

J.L. Elliott and D. Tesar

Dept. of Mech. Engrg., Univ. of Florida, Gainesville, FL., J. Engr. Indus., Trans. ASME, 99 (3), pp 715-722 (Aug 1977) 9 figs, 10 tables, 30 refs**Key Words:** Linkages, Mechanisms, Balancing

A method for the driving torque, shaking moment, and shaking force balancing is given as individual or combined problems for all of the four-link mechanisms: the four-bar, the slider-crank, the inverted slider-crank, and the oscillating block mechanism.

77-2127**A Numerical Method for the Dynamic Analysis of Mechanical Systems in Impact**

R.E. Beckett, K.C. Pan, and S.C. Chu

Gen. Thomas J. Rodman Lab., Rock Island Arsenal, Rock Island, IL., J. Engr. Indus., Trans. ASME, 99 (3), pp 665-673 (Aug 1977) 15 figs, 1 table, 16 refs**Key Words:** Mechanisms, Linkages, Dynamic response, Numerical analysis

A general procedure is developed for solving mechanism problems where intermittent separations and impacts can occur between mating parts. The numerical technique employed to solve the problem identifies the onset of separation and gives the behavior of the mechanism during separation and impact.

77-2128**Shape and Frequency Composition of Pulses From an Impact Pair**

R.G. Herbert and D.C. McWhannell

Dept. of Mech. Engrg., Univ. of Southampton, UK, J. Engr. Indus., Trans. ASME, 99 (3), pp 513-518 (Aug 1977) 10 figs, 9 refs**Key Words:** Impact pairs, Linkages, Mechanisms, Noise generation

With the need to improve the reliability and noise emissions from real mechanisms, an impact in the classical impact pair configuration is investigated. The impact pulse level and its frequency composition as possible sources of high-frequency energy in articulated systems is considered.

77-2129**Dynamic Response of a Cam-Actuated Mechanism with Pneumatic Coupling**

F.Y. Chen

Dept. of Mech. Engrg., Ohio Univ., Athens, OH, J. Engr. Indus., Trans. ASME, 99 (3), pp 598-603 (Aug 1977) 7 figs, 6 refs

Key Words: Cam followers, Pneumatic equipment, Stability, Dynamic response, Lumped parameter method

The dynamic characteristics of a cam-actuated system whose follower mass is coupled with a nonlinear pneumatic mechanism of hysteretic type are investigated using a lumped-parameter model. The dynamic response of the cam follower is obtained from the solution of the formulated system equation by the Krylov-Bogoliubov method of variation of parameters. The stability of the system is investigated.

77-2130

A Survey of the State of the Art of Cam System Dynamics

F.Y. Chen

Dept. of Mech. Engrg., Ohio Univ., Athens, OH 45701, Mech. and Mach. Theory, 12 (3), pp 201-224 (1977) 17 figs, 128 refs

Key Words: Cams, Dynamic properties

The primary goal of this report is to present a comprehensive survey of the state of knowledge on the kinematic and dynamic aspects of the cam driven mechanisms and systems. The kinematics deals with the geometry of the cam curve, its continuity, curvature and boundary conditions as well as the mathematical derivatives of the curve which govern the velocity and acceleration characteristics of the motion. The dynamic problem areas concern physical modeling, formulations of the equations of motion, solution techniques, presentation of system's responses and the influence of design parameters.

PIPES AND TUBES

77-2131

An Experimental Investigation of Flow in an Oscillating Pipe

M. Clamen and P. Minton

Dept. of Civil Engrg., Imperial College, London, UK, J. Fluid Mech., 81 (3), pp 421-431 (July 13, 1977) 8 figs, 13 refs

Key Words: Pipes, Fluid-induced excitation

The hydrogen-bubble technique has been used to measure the velocities of pulsating water flow in a rigid circular pipe. Mean flows with Reynolds numbers between 1275 and 2900 were superimposed on an oscillating flow produced by moving the pipe axially with simple harmonic motion. While the velocities in the oscillating boundary layers on the pipe wall were found to be close to those predicted by laminar

flow theory, at the higher Reynolds numbers the velocities near the center of the pipe were lower than those predicted and more uniformly distributed. The intermittency of the periodic bursts of turbulent motion at the higher Reynolds numbers was measured. At each mean-flow Reynolds number the turbulent intermittency of the flow was found to be a function of a single parameter: the harmonic-flow Reynolds number.

77-2132

Vibration of Tubes Conveying Fluids

V.A. Svetitsky

Moscow Higher Tech. School, Moscow, USSR, J. Acoust. Soc. Amer., 62 (3), pp 595-600 (Sept 1977) 4 figs, 18 refs

Key Words: Pipes (tubes), Fluid-induced excitation

General, nonlinear equations are derived for the vibration of rectilinear tubes conveying incompressible fluid. From these equations are obtained the equations for small vibrations. If values of tube frequencies and critical flow parameters are to be predicted accurately, the initial state of stress must be taken into account. A numerical example is considered.

77-2133

Bifurcations to Divergence and Flutter in Flow-Induced Oscillations: A Finite Dimensional Analysis

P.J. Holmes

Inst. of Sound and Vibration Res., Univ. of Southampton, Southampton SO9 5NH, UK, J. Sound Vib., 53 (4), pp 471-503 (1977) 16 figs, 35 refs

Key Words: Pipes (tubes), Flutter, Flow-induced excitation

The behavior of a pipe conveying fluid and a fluid loaded panel are studied from the viewpoint of differentiable dynamics. Non-linear terms are included. A general approach for solution is illustrated by analysis of two mode models of a pipe and of a panel and some important omissions in previous treatments of linear and undamped systems are discussed.

77-2134

A Preliminary Study of Flow and Acoustic Phenomena in Tube Banks

J.A. Fitzpatrick and I.S. Donaldson

Univ. of Glasgow, Glasgow, UK, ASME Paper No. 77-FE-7

Key Words: Tubes, Acoustic response, Wind tunnel tests

Experiments have been performed in a low-turbulence wind tunnel to investigate the effect of tube pitch to diameter ratios, depth of bank and Reynolds number on the parameters associated with resonant acoustic vibration in in-line tube banks.

77-1235

Experimental Data on the Natural Frequency of a Tubular Array

B.T. Lubin, K.H. Haslinger, A. Puri, and J. Goldberg
Combustion Engineering, Inc., Windsor, CT, ASME
Paper No. 77-FE-10

Key Words: Tubes, Natural frequency

Data from experiments on an array of tubes in water showed that the tubes vibrated over a range of frequencies centered about an isolated single tube frequency. The concept of a motion dependent hydrodynamic mass has been successfully used to explain the observed results.

77-2136

Exchanger Design Cuts Tube Vibration Failures

W.M. Small and R.K. Young
Phillips Petroleum Co., Bartlesville, OK, Oil and Gas
J., 75 (37), pp 77-80 (Sept 5, 1977) 5 figs, 1 table

Key Words: Tubes, Heat exchangers, Vibration reduction

Rod-baffle heat exchanger design is described which solves the problem of tube failures due to vibration and provides a low pressure drop across the bundle.

77-2137

Flow-Induced Vibrations of a Hydraulic Valve and Their Elimination

D.S. Weaver, F.A. Adubi, and N. Kouwen
McMaster Univ., Hamilton, Ontario, Canada, ASME
Paper No. 77-FE-24

Key Words: Hydraulic valves, Fluid-induced excitation

The flow-induced vibrations of a check valve with a spring damper to prevent slamming have been studied experimentally. Both prototype and two-dimensional model experiments were conducted to develop an understanding of the mechanism of self-excitation. The phenomenon is shown to be caused by the high rate of change of discharge at small angles of valve opening and the hysteretic hydrodynamic loading resulting from fluid inertia.

PLATES AND SHELLS

(Also see Nos. 2062, 2071)

77-2138

An Analogy Between Free Vibration of a Plate and of a Particle of Mass

Z. Celep

Faculty of Engrg. and Architecture, Technical Univ.,
Istanbul, Turkey, J. Sound Vib., 53 (3), pp 323-331
(Aug 8, 1977) 5 figs, 10 refs

Key Words: Plates, Free vibration, Flexural vibration

In this paper, the free flexural vibration of an elastic circular thin plate with an initial imperfection is investigated. Approximate solution of the problem for the fundamental frequency of vibration, of large amplitude and with the plate imperfection, leads to a nonlinear ordinary differential equation with time as the independent variable.

77-2139

Measurement of Mechanical Vibration Damping in Orthotropic, Composite and Isotropic Plates Based on a Continuous System Analysis

N. Basavanhally and R.D. Marangoni
Dept. of Mech. Engrg., Univ. pf Pittsburgh, Pittsburgh,
PA 15261, Intl. J. Solids Struc., 13 (8), pp
669-707 (1977) 8 figs, 9 refs

Key Words: Plates, Vibration damping, Measurement techniques

The problem of free and forced transverse vibration of an orthotropic, composite, and isotropic thin square plates with uniformly distributed damping and simply supported boundary conditions has been solved, using a modal expansion technique. A load of the type $P_0 \cos \omega t$ applied at the center of plate has been considered and the phase angle between the forcing function and the vibration response at the center, as a function of the forcing frequency and the damping parameter determined. This theoretical relationship together with the experimentally measured phase angle between the applied mechanical forcing and the resulting vibration response at various forcing frequencies was used to determine an equivalent viscous damping parameter. This technique has been found to be particularly useful for the measurement and comparison of the relative damping in composite or orthotropic materials. Also, a theoretical relation for the energy loss due to viscous damping in vibrating plates has been developed and the theoretical energy loss at various frequencies has been compared with the experimentally measured energy loss at the same frequencies. Typical damping results are presented for aluminum, steel and aluminum/graphite-fiber composite materials.

77-2140

Variable Order Finite Elements for Plate Vibration

J.R. Hutchinson and J.J. Benitou

Dept. of Civil Engrg., Univ. of Calif., Davis, CA, ASCE J. Engr. Mech. Div., 103 (EM5), pp 779-787 (Oct 1977) 3 tables, 12 refs

Key Words: Plates, Finite element technique, Natural frequencies, Mode shapes

Rectangular finite elements with a variable number of degrees-of-freedom per element are developed for thin elastic plates. The displacement field for the element is described by a fixed series of polynomial terms plus a variable number of trigonometric terms.

Stability of a rectangular elastic plate is investigated by means of a Liapunov's Second Method. It is assumed that the plate is subjected to tangential follower forces which are parallel to one edge of the plate, that the plate has internal viscous damping, and that it is simply supported and/or clamped along its contour. The main result is that only for sufficiently large damping, stability is ensured for reasonably large follower forces.

77-2141

Loss Factor for a Rectangular Plate of Parabolic Thickness Variation

S.P. Nigam, G.K. Grover, and S. Lal

Mech. Engrg., Government Engrg. College, Jabalpur, India, J. Engr. Indus., Trans. ASME, 99 (3), pp 799-801 (Aug 1977) 2 tables, 4 refs

Key Words: Rectangular plates, Variable cross section, Fundamental mode, Internal damping

The importance of the internal damping and of the evaluation of the fundamental mode loss factor of structural members subjected to multiaxial stress system is well known. It appears that little work has been done on vibrations of rectangular plates of variable thickness, though such cases are of interest in the aeronautical field since they approximate to wing sections. In the present work, the fundamental mode loss factors for a simply supported rectangular plate with parabolic thickness variation in X direction have been evaluated for different combinations of the aspect ratios and the taper parameters. An approximate relationship has been obtained which correlates the loss factor for the plate of variable thickness with that of a plate of uniform thickness.

77-2143

Large Amplitude Radial Oscillations of Layered Thick-Walled Cylindrical Shells

A. Ertepinar

Dept. of Engrg. Sciences, Middle East Technical Univ., Ankara, Turkey, Intl. J. Solids Struc., 13 (8), pp 717-723 (1977) 4 figs, 1 table, 9 refs

Key Words: Cylindrical shells, Oscillation

Finite breathing motions of multi-layered, long, circular cylindrical shells of arbitrary wall thickness are investigated on the basis of the theory of large elastic deformations. The materials of the layers are assumed to be isotropic, elastic, homogeneous and incompressible. The governing non-linear ordinary differential equation is solved partially to give the frequencies of oscillations in an integral form. An approximate solution technique based on Galerkin's orthogonalization process is also formulated to give complete solutions. A tube consisting of two layers of new-Hookean materials is solved both by exact and approximate methods. An excellent agreement is observed between the two sets of results.

77-2144

Axially Symmetric Vibrations of Finite Cylindrical Shells of Various Wall Thicknesses - II

J. Chandra and R. Kumar

Systems Engrg., Div., Defence Science Lab., Delhi-110054, India, Acustica, 38 (1), pp 24-29 (July 1977) 9 figs, 2 refs

Key Words: Cylindrical shells, Resonant frequency, Axisymmetric vibration

Using the exact three-dimensional equations of linear elasticity, the vibrational characteristics of circular cylindrical shells of various wall thicknesses and finite length have been studied. The motion of the shell is assumed to be axially symmetric but anti-symmetric about its central plane. The stress-free conditions on the lateral surface of the shell have been satisfied exactly and the real, imaginary and complex branches of the dispersion spectra have been superposed to satisfy the stress-free conditions at the flat ends of the shell to a good degree of accuracy. The aspect ratio curves, the residual stresses at the flat ends and the displacements have been given for various wall thicknesses.

77-2142

Stability of Elastic Plates via Liapunov's Second Method

H.H.E. Leipholz

Solid Mech. Div., Faculty of Engrg., Univ of Waterloo, Waterloo, Ontario N2L 3G1, Canada, Ing. Arch., 45 (5/6), pp 337-345 (1976) 4 figs, 3 refs

Sponsored by the National Res. Council of Canada

Key Words: Rectangular plates, Follower forces, Liapunov's method

77-2145

Analysis of a Cylindrical Shell Vibrating in a Cylindrical Fluid Region

H. Chung, P. Turula, T.M. Mulcahy, and J.A. Jendrzejczyk

Components Tech. Div. Argonne National Lab., IL,
Rept. No. ANL-76-48, 24 pp (Aug 1976)
N77-26544

Key Words: Cylindrical shells, Nuclear reactor components, Natural frequencies, Mode shapes, Computer programs

Analytical and experimental methods are presented for evaluating the vibration characteristics of cylindrical shells such as the thermal liner of the Fast Flux Test Facility (FFTF) reactor vessel. The NASTRAN computer program is used to calculate the natural frequencies, mode shapes, and response to a harmonic loading of a thin, circular cylindrical shell situated inside a fluid-filled rigid circular cylinder. Solutions in a vacuum are verified with an exact solution method and the SAP IV computer code. Comparisons between analysis and experiment are made, and the accuracy and utility of the fluid-solid interaction package of NASTRAN is assessed.

77-2146

Vibrations of Prolate Spheroidal Shells of Constant Thickness

C.B. Burroughs

Ph.D. Thesis, The Catholic Univ. of America, 35 pp (1977)
UM 77-17,514

Key Words: Spherical shells, Fluid-induced excitation, Transverse shear deformation effects, Rotatory inertia effects

The general displacement-equilibrium equations, which include the effects of transverse shear and rotary inertia, have been derived for a fluid-loaded prolate spheroidal shell of constant thickness subject to an harmonically time-varying, arbitrary spatially distributed force normal to the shell surface. The solution is formulated for the axisymmetric motion of a shell that is immersed in an inviscid fluid of infinite extent. The approximate solutions for the two nontorsional displacements of the shell middle surface and the nontorsional rotation of the shell cross-section are obtained by using an extension of Galerkin's variational method developed by Chi and Magrab.

77-2147

Vibration of Complex Structures by Matching Spatially Dependent Boundary Conditions of Classical Solutions. Specifically Vibration Characteristics of

Hollow Symmetrical Blades Based on Thin Shell Theory

A.M. Al-Jumaily

Ph.D. Thesis, The Ohio State Univ., 230 pp (1977)
UM 77-17,072

Key Words: Blades, Shells, Plates, Beams, Resonant frequencies, Mode shapes

The mathematical formulation and solution methods for dynamics problems of continuous structures composed of beam, plate, and shell elements are investigated by developing and using the Matching of Continuous Boundary Conditions Technique. This technique results in a closed form functional solution for the resonant frequencies and corresponding mode shapes of the composite structure. A hollow symmetrical turbomachinery blade is used to illustrate the general method. The blade is composed of two-co-linear open profile circular cylindrical shell elements connected at their straight edges. Experimental investigations are performed to support the results of the theories. In the course of formulating the blade problem, two new simplified shell solution techniques are introduced. One is based on Yu's assumptions for shells with small radius to length ratios; the second theory is derived from basic principles based on different assumptions gathered from the literature. The results of using the simplified shell solution technique, the Matching of Continuous Boundary Conditions method, and the experimental investigations are compared. Other methods of solution for dynamic problems of continuous structures, such as the Point Matching Technique, are investigated.

77-2148

The Effect of Creep Deformation on the Vibration and Stability Characteristics of Axisymmetric Thin Shells

A.P. Gelman

Ph.D. Thesis, Univ. of Southern California (1977)

Key Words: Shells, Natural frequencies, Computer programs, Stiffness methods

An analysis and a computer program have been developed for calculating the changes in the natural frequencies of axisymmetric thin shells when they are subjected to axisymmetric loads and are permitted unrestricted creep. The method of solution is an extension of the direct stiffness method. The shell is replaced by a system of discrete finite elements consisting of conical frusta; these elements are interconnected along circumferential nodal circles. The dynamical equations of equilibrium are obtained from the principles of minimum potential energy. The Sanders nonlinear strain displacement relations are utilized to obtain a linear stiffness matrix, a stress dependent geometric stiffness matrix, a nonlinear large displacement matrix, and a consistent mass matrix.

STRUCTURAL

77-2149

Earthquake Response of Coupled Shear Wall Buildings

T. Srichatrapimuk

Ph.D. Thesis, Univ. of California, Berkeley, 122 pp (1976)

UM 77-15,866

Key Words: Buildings, Walls, Earthquake response

An efficient analytical technique for determining linear and nonlinear response of coupled shear wall structures is developed. Walls are assumed to be nonyielding with all inelastic action confined to coupling beams. Structural displacements are then represented as a linear combination of the first few natural mode shapes in both lateral and longitudinal (vertical) vibration of individual walls which are treated as independent cantilevers. The effectiveness and flexibility of this general approach in reducing the number of degrees of freedom are demonstrated. The analytical technique is implemented in earthquake response analyses of two coupled shear wall systems; analytical results are then correlated with observations of earthquake damage in these structures. The earthquake response of coupled shear walls is then interpreted, and design considerations for efficient earthquake resistant shear wall systems are suggested.

77-2150

Air Blast Effects on Concrete Walls

C.A. Kot and P. Turula

Argonne National Lab., IL, Rept. No. ANL-CT-76-50, 67 pp (July 1976)

N77-26540

Key Words: Walls, Concrete construction, Blast effects

Estimates are obtained both for the spalling of the back-face of the concrete wall and for the overall wall response produced by the total impulsive load of the air blast. Assuming elastic wave propagation in the concrete wall, it is found that as spall thickness increases, the spall velocity decreases. This holds for normal as well as oblique wave incidence on the back-face of the wall. Therefore, for debris which has significant mass, the ejection velocity produced by spalling action alone is quite moderate. Plastic yield-line analysis of the wall segment subjected to the impulsive loading of the air blast indicates that for sufficiently large explosions substantial displacements and peak velocities can occur in typical shield walls.

77-2151

Seismic Response of a Periodic Array of Structures

H. Murakami and J.E. Luco

Dept. of Applied Mech. & Engrg. Sci., Univ. of Calif. at San Diego, La Jolla, CA, ASCE J. Engr. Mech. Div., 103 (EM5), pp 965-977 (Oct 1977) 6 figs, 12 refs

Key Words: Walls, Buildings, Earthquake response

A simplified two-dimensional model of the dynamic interaction, through the soil, among adjacent structures in a densely built area is presented. The model consists of an infinite number of identical parallel infinitely long shear walls placed on equally spaced rigid semi-cylindrical foundations. The steady-state response of the shear walls to obliquely incident plane SH waves is evaluated and compared with the response of an isolated structure.

SYSTEMS

ABSORBER

(Also see No. 2088)

77-2152

Design of Viscous Torsional Vibration Absorbers (Auslegung von Viskositätsdreh-Schwingungsdämpfern)

R. Mehner

Tech. Univ. Dresden, German Democratic Republic, Maschinenbautechnik, 27 (7), pp 326-329 (July 1977) 8 figs, 5 refs
(In German)

Key Words: Optimization, Vibration absorbers

An exact method for the optimization of vibration absorbers is obtained from the relationship of single mass systems with the viscosity torsional vibration absorbers. The method is based on electronic data processing.

NOISE REDUCTION

(Also see Nos. 2061, 2163, 2194)

77-2153

Machinery Noise Reduction. Correct Design Improves Efficiency (Lärmabschirmungen an Maschinen. Richtiges Gestalten erhöht die Wirksamkeit.)

J. Thoma

Techn. Rdschau (Bern), 68 (38), p 33 (1976) 1 fig, 3 refs

Key Words: Machinery noise, Noise reduction

The topics discussed are active and passive measures, simplified physics of noise, reflection, absorption and transmission of noise, noise amplification by means of reflection of sound in protective housing, harmful effects of small holes, absorption and stiffening for increasing the effectiveness of housing.

77-2154

Reducing Machinery Noise

R.L. Hershey

Booz, Allen & Hamilton, Inc., Indus. Res., 19 (9), pp 118-121 (Sept 1977) 6 refs

Key Words: Machinery noise, Noise reduction, Regulations

Considerable research has been devoted to reducing the noise from industrial machinery, such as circular saws, punch presses, textile spinning frames, and typewriters. This article describes some of the research areas and the regulations that have provided impetus toward quieting these machines.

77-2155

Systems for Noise and Vibration Control

W.E. Purcell

S/V, Sound Vib., 11 (8), pp 4-30 (Aug 1977)

Key Words: Noise reduction, Acoustic absorption, Noise barriers, Vibration control

Systems for noise and vibration control are finished products or components generally designed for specific purposes. For his discussion the author classifies such systems into: silencers, sound absorptive systems, sound barrier systems, and vibration/shock control systems.

77-2156

Acoustical Scale Model Study of the Attenuation of Sound by Wide Barriers

E.S. Ivey and G.A. Russell

Dept. of Physics, Smith College, Northampton, MA 01060, J. Acoust. Soc. Amer., 62 (3), pp 601-606 (Sept 1977) 8 figs, 15 refs

Key Words: Noise barriers, Acoustic attenuation, Model testing

Acoustical scale model experiments carried out with building-size barriers are described. The results of experiments conducted with the barrier in a free field and on a reflecting surface are presented. The free field measurements are compared to several theoretical models and discrepancies between the theoretical and experimental results are discussed. Also presented is a simple expression which relates the excess attenuation obtained with the barrier situated on the ground to that of the same barrier in the free field. This expression predicts excess attenuations which agree quite closely with those actually measured in the scale model experiments.

77-2157

OSHA and the Noise of Pneumatic Systems

R.C. Potter

Bolt Beranek and Newman, Inc., Cambridge, MA, ASME Paper No. 77-DE-49

Key Words: Pneumatic equipment, Noise reduction

Pneumatic systems produce high-level sounds in that part of the frequency spectrum that has the most influence on human hearing. OSHA requires that the hearing of individual workers be protected, and it is often the pneumatics of a machine that will control the sound levels received. Descriptions are given of the noise produced by the compressors that supply the air, the pipes and valves that transmit and control the air, and the devices, mechanisms, and tools that use the air. Methods are discussed for reducing the noise, and it is concluded that both management and employees will benefit from consideration of the problem of pneumatic system noise in present plants and in the design of future installations.

AIRCRAFT

(Also see No. 2197)

77-2158

Supersonic Jet Exhaust Noise Investigation, Volume IV. Acoustic Far-Field/Near-Field Data Report

P.R. Knott and J.F. Brausch

Aircraft Engine Group, General Electric Co., Cincinnati, OH, Rept. No. R74-AEG452-Vol-4, AFAPL-TR-76-68-Vol-4, 504 pp (July 1976)

AD-A040 894/8GA

Key Words: Jet noise, Aircraft noise

This report is an acoustic data report presenting a series of parametric acoustic far-field and near-field results for subsonic and supersonic heated flow conditions for a simple conical nozzle (thin lip and thick lip) and a convergent-divergent nozzle at design and off-design conditions.

77-2159

Recommended Procedures for Measuring Aircraft Noise and Associated Parameters

A.H. Marsh

DyTec Engrg., Inc., Huntington Beach, CA., Rept. No. NASA-CR-145187, 164 pp (Apr 1977) refs N77-25912
N77-25912

Key Words: Aircraft noise, Noise measurement

Procedures are recommended for obtaining experimental values of aircraft flyover noise levels (and associated parameters). Specific recommendations are made for test criteria, instrumentation performance requirements, data-acquisition procedures, and test operations. The recommendations are based on state-of-the-art measurement capabilities available in 1976 and are consistent with the measurement objectives of the NASA Aircraft Noise Prediction Program. The recommendations are applicable to measurements of the noise produced by an airplane flying subsonically over (or past) microphones located near the surface of the ground. Aircraft types covered by the recommendations are fixed-wing airplanes powered by turbojet or turbofan engines and using conventional aerodynamic means for takeoff and landing. Various assumptions with respect to subsequent data processing and analysis were made (and are described) and the recommended measurement procedures are compatible with the assumptions. Some areas where additional research is needed relative to aircraft flyover noise measurement techniques are also discussed.

77-2160

Problems in Predicting Aircraft Noise Exposure

A.H. Odell

Port Authority of New York and New Jersey, One World Trade Ctr. 65S, New York, NY 10048, Noise Control Engr., 9 (1), pp 32-37 (July/Aug 1977)
9 figs, 21 refs

Key Words: Aircraft noise, Noise prediction, Human response

For more than twenty years, the aviation industry has tried to develop a single universal rating method which would accurately describe the noise produced by aircraft operations in terms of the subjective reaction of the exposed population. Some of the basic assumptions involved in this procedure are examined by the author. Also offered are suggestions for improvement in the methodology and potential areas of study.

77-2161

On the Growth Rate of Bending Induced Edge Cracks in Acoustically Excited Panels

K.P. Byrne

Dept. of Mech. and Industrial Engrg., Univ. of New South Wales, Kensington, NSW 2033, Australia, J. Sound Vib., 53 (4), pp 505-528 (1977) 16 figs, 1 table, 9 refs

Key Words: Aircraft, Acoustic excitation, Acoustic fatigue

The emphasis of the work described in this paper is on examining the growth rate of edge cracks in acoustically excited panels. A single panel with an edge crack is considered and this structural element is modelled as a flat plate clamped on three edges and part of the fourth. The crack is represented by the unclamped part of the fourth edge. Fracture mechanics principles are used to predict the crack growth rates associated with the first two modes of vibration of the edge cracked panel. The crack tip stress intensity factors associated with these panel modes are estimated by a technique based on finding the nominal bending stresses at the crack tips. The nominal bending stresses are in turn found from mode shapes determined by the Rayleigh Principle. The validity of the various assumptions is assessed by comparing the predicted crack growth rates with measured growth rates in panels representative of those used in aircraft construction.

77-2162

Non-Linear Effects in Aircraft Ground and Flight Vibration Tests

G. Haidl

Messerschmitt-Boelkow-Blohm G.m.b.H., Ottobrunn, Fed. Rep. Germany, Rept. No. MBB-UFE-1273-0, 16 pp (Sept 16, 1976) refs
N77-25153

Key Words: Aircraft, Resonance tests, Vibration tests, Flutter

Examples of nonlinear vibration behavior in ground resonance tests of an aircraft are shown. Model tests for a simplified system with nonlinear properties were performed to study the effects of friction and backlash with respect to ground resonance test and flight flutter test. With symmetric and asymmetric nonlinear stiffness characteristics effects of amplitude dependent frequencies, mode coupling, mode asymmetries, and the consequences in parameter identification in vibration tests are pointed out and discussed. In case of flutter critical modes the problems of apparent damping caused by nonlinear system properties are shown, and recommendations are given to reach a representative flutter clearance with respect to this nonlinear system behavior.

77-2163

Supersonic Transport Noise Reduction Technology Program - Phase II. Volume I

S.B. Kazin, E.J. Stringas, J.T. Blozy, V.L. Doyle, and R.B. Mishler
Aircraft Engine Group, General Electric Co., Cincinnati, OH, Rept. No. R75AEG362-Vol-1, FAA-SS-73-29-1, 478 pp (Sept 1975)
AD-B010 468/7GA

Key Words: Supersonic aircraft, Noise reduction

The Supersonic Transport Noise Reduction Technology Program, sponsored by the Federal Aviation Administration, was conducted as a follow-on effort after cancellation of the SST Program to finalize selected noise technology areas and summarize results of the SST Program. The overall program objective was to provide additional acoustic technology necessary to design high speed aircraft systems, recognizing future acceptable noise levels. General Electric's effort was divided into the acoustic technology areas of jet noise reduction, turbomachinery noise reduction, and aircraft system integration. Jet noise reduction technology work was achieved through analytical studies, model tests, and J79 engine tests. Selected suppression systems identified during the SST Program were further refined (multispoke/chute suppressors or annular plug nozzles). Novel advanced concepts of suppression were identified, and extensive aerodynamic (static and wind-on) performance tests and hot-jet acoustic tests were performed.

77-2164

Airframe, Wing, and Tail Aerodynamic Characteristics of a 1/6-Scale Model of the Rotor Systems Research Aircraft with the Rotors Removed

R.E. Mineck and C.E. Freeman
Army Air Mobility Res. and Dev. Lab., Hampton,

VA, Rept. No. NASA-TN-D-8456, 141 pp (May 1977)

N77-26082

Key Words: Aircraft, Helicopters, Airframes, Aircraft wings, Wind tunnel tests

A wind-tunnel investigation was conducted to determine the aerodynamic characteristics of the rotor systems research aircraft (RSRA) as the helicopter and the compound helicopter with the rotors removed. Data were obtained over ranges of angles of attack and angle of sideslip. Results are presented for the total loads on the airframe as well as the loads on the wing and the tail.

77-2165

Treatment of the Nonlinear Vibration of a Variable Sweep Aircraft Wing with its Drive Using a Simplified Wing Model (Behandlung des nichtlinearen Schwingungsverhaltens eines schwenkbaren Flugzeugflügels mit seinem Verstellantrieb mittels eines vereinfachten Schwingungsmodells)

B. Schoen
Unternehmensbereich Flugzeuge-Entwicklung, Messerschmitt-Boelkow-Blohm G.m.b.H., Ottobrunn, W. Germany, Rept. No. MBB-UFE-1191(0), 155 pp (Aug 1, 1975)
(In Georgian)
N77-26156

Key Words: Aircraft wings, Vibration response, Mathematical models

A wing vibration model was constructed to investigate the vibration behavior of a variable sweep wing with its pivot drive. The model provides for simulation of the clearance, the static friction, and damping proportional to velocity. The physical vibration behavior was investigated by variation of these parameters. The complex phenomenon was also studied theoretically by approximation solutions, and the dependence on parameter variations indicated. Experimental and theoretical results are combined to provide a complete picture of the vibration phenomenon.

77-2166

Flutter Analysis of an All-Movable Horizontal Tail with Geared Elevator on a Supersonic Transport

J.L. Stelma
Boeing Commercial Airplane Co., Seattle, WA, Rept. No. D6-60293, FAA-SS-73-16, 60 pp (June 1974)
AD-B000 285/7GA

Key Words: Flutter, Supersonic aircraft

This document presents symmetric flutter analyses conducted on the all-movable horizontal tail of the Boeing-designed SST. Interaction effects on flutter speed that are produced by the wing, fuselage, control systems and elevator gear ratio are included. Failure conditions of the horizontal-tail actuators are covered.

The natural mechanical modes of vibration in bending (vertical and lateral) and torsion are assumed known, and the response of each of these with postulated negligible aero-dynamic coupling between modes, is calculated. Some examples are then given of the calculated vertical and torsional buffeting responses of a flexible long-span bridge (Golden Gate type) and a stiff, medium-span type (Sitka Harbor). The wind velocity range covered is 60 mph to 90 mph (27 m/s to 40 m/s).

77-2167

A Low Speed Model Analysis and Demonstration of Active Control Systems for Rigid-Body and Flexible Mode Stability

R.A. Gregory, A.D. Ryneveld, and R.S. Imes
Boeing Commercial Airplane Co., Seattle, WA, Rept. No. D6-60295, FAA-SS-73-18, 203 pp (June 1974) AD-B000 286/5GA

Key Words: Supersonic aircraft, Flutter, Wind tunnel tests, Stability analysis

An existing low-speed SST flutter model was modified to include two hydraulic aileron control systems and a horizontal stabilizer system. Wing mode flutter suppression systems were analyzed and wing tunnel tested, using wing strain gages and the aileron systems in the active control feedback loops. Rigid-body stability systems were theoretically analyzed and experimentally synthesized using body-mounted sensors. Variable rigid-body stability was achieved through a remote-transfer water ballast system. The results of parallel analysis and wind tunnel tests, the methods of approach, the problems encountered, and a list of recommendations for the advancement of the active controls technology are reported in this document.

BRIDGES

(Also see No. 2110)

77-2168

Motion of Suspended Bridge Spans under Gusty Wind

R.H. Scanlan and R.H. Gade
ASCE J. Struc. Div., 103 (ST9), pp 1867-1883 (Sept 1977) 5 figs, 47 refs, 5 tables

Key Words: Suspension bridges, Wind-induced excitation

The buffeting response of suspended-span bridges can be calculated if certain wind-tunnel section model data, plus wind spectral information, are provided. The needed wind-tunnel data are the self-excited aerodynamic (flutter) coefficients. The meteorological data required are vertical and horizontal gust spectra of the natural wind at the bridge site.

77-2169

Effects of Uniform and Non-Uniform Seismic Disturbances on a Long Multi-Span Highway Bridge

R.E. Hamati
Ph.D. Thesis, Univ. of Calif., Berkeley, 397 pp (1976) UM 77-15,710

Key Words: Bridges, Seismic design

Criteria were developed for the seismic design of a long multi-span highway bridge. The criteria are for requirements of seismic strength to resist inertia effects, and provisions for sufficient ductility to absorb the displacements and deformations caused by uniform and non-uniform distributions of ground motions. Criteria were also developed for determining the ductilities and capacities of elements of the bridge to absorb the maximum relative displacements that may be caused by residual deformations of the soils. In developing the criteria, various parameters were considered. Among the parameters are those related to bridge types, articulations, soil conditions, and spatial distributions of ground motions. The effects of soil-structure interaction are included.

BUILDING

(Also see Nos. 2083, 2088, 2149, 2151)

77-2170

Inelastic Earthquake Response of Three-Dimensional Buildings

R. Guendelman-Israel
Ph.D. Thesis, Univ. of Calif., Berkeley, 130 pp (1976) UM 77-15,705

Key Words: Buildings, Earthquake response, Computer programs

A computational procedure and computer program for the inelastic dynamic response analysis of three-dimensional buildings of essentially arbitrary configuration is described. The building is idealized as a series of independent plane substructures interconnected by horizontal rigid diaphragms. Each substructure can be of arbitrary geometry and include structural elements of a variety of types.

77-2171**Inelastic Response to Site-Modified Ground Motions**

R.V. Whitman and J.N. Protonotarios

Mass. Inst. of Tech., Cambridge, MA, ASCE J. Geotech. Engr. Div., 103 (GT10), pp 1037-1053 (Oct 1977) 16 figs, 1 table, 12 refs**Key Words:** Buildings, Earthquake response

A building with a period equal to that of a site may be more susceptible to yielding during a moderate earthquake, but the larger yielding during a major earthquake is much the same as for a building having a different period. This conclusion results from analyzing one-degree-of-freedom, elastoplastic structures using ground motions (both real and calculated) whose elastic response spectra have peaks attributable to site conditions. Inelastic response spectra for site-modified motions do not show pronounced peaks at the period of the site; rather, they are as "smooth" as inelastic spectra computed from motions unaffected by site conditions. Inelastic spectra for design may be based upon the same ratios of spectral acceleration to peak acceleration and spectral velocity to peak velocity as for normal motions. Thus, the amount by which a site modifies peak acceleration and peak velocity is important, and the period of a site is not significant by itself.

77-2172**Review of Literature on Earthquake Damage to Single-Family Masonry Dwellings**

R.D. Benson

Applied Tech. Council, Palo Alto, CA, 31 pp (Apr 29, 1977)

PB-267 947/0GA

Key Words: Earthquake damage, Buildings, Masonry, Reviews

The report contains a review and evaluation of information concerning the behavior of single-family masonry dwellings in Zone 2 earthquake areas of the United States (1973 Uniform Building Code classification). In general, reinforced masonry has exhibited satisfactory performance, sustaining little or no damage in moderate earthquakes. Reported damage is often associated with poor workmanship/inspection. Unreinforced masonry (old and new) and masonry chimneys have exhibited poor performance. Available data has been found to be limited and general in nature.

FOUNDATIONS AND EARTH

(See Nos. 2084, 2106)

HELICOPTERS

(Also see No. 2164)

77-2173**Aeroelastic Stability of Complete Rotors with Application to a Teetering Rotor in Forward Flight**

J. Shamie and P. Friedmann

Mechanics and Structures Dept., School of Engrg. and Applied Science, Univ. of Calif., Los Angeles 90024, J. Sound Vib., 53 (4), pp 559-584 (1977) 12 figs, 23 refs**Key Words:** Helicopter rotors, Dynamic stability

The derivation of a set of non-linear coupled flap-lag-torsion equations of motion for moderately large deflections of an elastic, two-bladed teetering helicopter rotor in forward flight is concisely outlined.

77-2174**Effect of Production Modifications to Rear of Westland Lynx Rotor Blade on Sectional Aerodynamic Characteristics**

P.G. Wilby

Aerodynamics Dept., Royal Aircraft Establishment, Farnborough, UK, Rept. No. ARC-CP-1362, RAE-TR-73043; ARC-34835, 21 pp (1977) refs N77-25101

Key Words: Helicopter rotors, Rotary wings, Aerodynamic response

The RAE (NPL) 9615 airfoil was accepted, on the basis of wind tunnel tests, as the basic blade section for the Westland WG 13 Lynx helicopter rotor; however, production methods necessitated a modification to the rear profile of the blades which was considered sufficient to produce changes in the aerodynamic characteristics of the airfoil. Thus, the modified profile was tested in the wind tunnel and the experimental data compared with those for the original profile.

77-2175**Application of System Identification to Analytic Rotor Modeling from Simulated and Wind Tunnel Dynamic Test Data, Part 2**

K.H. Hohenemser and D. Banerjee

Dept. of Mech. Engrg., Washington Univ., St. Louis, MO, Rept. No. NASA-CR-152023, 194 pp (June 1977)

N77-26078

Key Words: Helicopters, Aircraft, Parameter identification, Rotors, Mathematical models

An introduction to aircraft state and parameter identification methods is presented. A simplified form of the maximum likelihood method is selected to extract analytical aeroelastic rotor models from simulated and dynamic wind tunnel test results for accelerated cyclic pitch stirring excitation. The dynamic inflow characteristics for forward flight conditions from the blade flapping responses without direct inflow measurements were examined.

obtainable. Methods to obtain the vehicle sprung mass and sprung mass moments of inertia are available; however, a simplified method to obtain the vehicle suspension spring rates, damping characteristics, and the unsprung mass inertia properties is needed. The technique that was developed in this thesis to obtain these suspension parameters requires a test of short duration, less than three seconds, and avoids vehicle disassembly. The parameters are identified from suspension force and displacement data, eliminating the need for complex calculations using detailed information concerning the characteristics and placement of each of the many components making up the suspension.

HUMAN

(Also see No. 2087)

77-2176

Hand-Arm Vibration Part II: Vibrational Responses of the Human Hand

J.W. Mishoe and C.W. Suggs

Agricultural Research and Education Ctr., Dept. of Agricultural Engrg., Univ. of Florida, Belle Glade 33430, J. Sound Vib., 53 (4), pp 545-558 (1977)
14 figs, 6 refs

Key Words: Human hand, Vibration response, Mathematical models, Mechanical impedance

When vibration is applied to the hand in the vertical (dorsal-to-ventral) and transverse direction, the hand arm system can be modeled by a three-mass model with each of the masses connected by a parallel spring and damper. For vibration input directed into the long axis of the forearm the model requires an additional parallel spring and damper to connect the last mass to an infinite base.

ISOLATION

77-2177

Equation Error Identification of Vehicle Suspension Parameters

D.M. Brueck

Ph.D. Thesis, Purdue, Univ., 200 pp (1976)
UM 77-15,384

Key Words: Suspension systems (vehicles), Parameter identification

A simplified method for the identification of vehicle suspension parameters is developed. Increased use of computer simulations in the design, development, and testing of vehicles requires that the various vehicle parameters be easily

MECHANICAL

77-2178

Active Electromagnetic Vibration Control in Rotating Discs

R.W. Ellis

Ph.D. Thesis, Univ. of Calif., Berkeley, 81 pp (1976)
UM 77-15,673

Key Words: Disks, Rotating structures, Saws, Vibration control

This thesis introduces a promising new technique for improving saw performance using an electronic feedback control system. The system consists of a non-contacting position sensor placed alongside the lateral surface of the saw, some control circuitry, and a pair of electromagnets placed alongside the saw, one on each side. The position sensor measures deviations from a normal undeflected condition and the control has produced significantly increased lateral stiffness and vibration damping characteristics in laboratory experiments, and it shows every indication of proving applicable to production situations.

METAL WORKING AND FORMING

77-2179

A Stability Analysis of Single-Point Machining with Varying Spindle Speed

J.S. Sexton, R.D. Milne, and B.J. Stone

Dept. of Mech. Engrg., Univ. of Bristol, Queens Bidg., Univ. Walk, Bristol BS8 1TR, UK, Appl. Math. Modeling, 1 (6), pp 310-318 (Sept 1977)
8 figs, 1 table, 8 refs

Key Words: Machine tools, Stability analysis, Chatter

The rate at which metal can be removed by a machine tool is often limited by the onset of an instability commonly

called 'chatter.' It has been suggested that greater widths of cut could be achieved without chatter on a given machine by modulating the spindle speed continuously. A stability analysis is presented which gives, for any mean spindle rotation speed and degree of modulation, the limiting width of cut for chatter-free cutting.

77-2180

Study on Optimum Design of Machine Structures with Respect to Dynamic Characteristics (Approach to Optimum Design of Machine Tool Structures with Respect to Regenerative Chatter)

M. Yoshimura

Faculty of Engrg., Kyoto Univ., Yoshida Sakyo-ku, Kyoto, Japan, Bull. JSME, 20 (145), pp 811-818 (July 1977) 10 figs, 3 tables, 5 refs

Key Words: Machine tools, Chatter

In order to attain dynamically optimum design of machine tools which would have minimum chance of machining chatter, an approach based on energy balances of a mathematical system at the resonance is developed and analyzed theoretically. This method aims that the maximum compliance of the tool-work relative displacement in the direction normal to cut across all frequency ranges. Using the computer simulations of machine tool structures, modal flexibilities are computed, by the magnitude of which the chance of regenerative chatter is judged.

77-2181

Identification and Active Adaptive Control of Chatter in Single-Point Machining Operations (Vol. I and II)

K. Srinivasan

Ph.D. Thesis, Purdue Univ., 883 pp (1976)
UM 77-15,476

Key Words: Machine tools, Chatter

Three areas of relevance to the active control of machine-tool chatter are considered in this thesis: Identification of machining system dynamics; controller design for machining systems; identification and controller adaptation for traverse machining operations.

77-2182

A New Approach to the Analysis of Machine-Tool System Stability under Working Conditions

F.A. Burney, S.M. Pandit, and S.M. Wu

Mech. Engrg. Dept., Univ. of Wisconsin, Madison,

J. Engr. Indus., Trans. ASME, 99 (3), pp 585-590 (Aug 1977) 6 figs, 2 tables, 20 refs

Key Words: Machine tools, Stability, Cutting, Mathematical models

A new stochastic approach is developed in this paper for analyzing the machine-tool system stability under working conditions. Mathematical models are fitted to the relative longitudinal cutter-workpiece displacement data recorded under different cutting conditions during the face-milling operation on a milling machine. The stability of the system is judged from the characteristic roots of these models. The variation in stability is examined versus both the cutting speed and the feed, and good results are obtained. It is shown that not only the dynamic but also the static stability can be ascertained. Furthermore, the stability of subsystems can also be determined. The significance of these results is discussed with special reference to on-line chatter control.

PUMPS, TURBINES, FANS, COMPRESSORS

(Also see No. 2157)

77-2183

Solve Vertical Pump Vibration Problems

R.J. Meyer

Industrial Pump Div., Allis-Chalmers Corp., Cincinnati, OH, Hydrocarbon Processing, 56 (8), pp 145-149 (Aug 1977) 6 figs

Key Words: Pumps, Vibration monitoring

Because of their long, slender structure, vertical pumps can have severe vibration problems. Possible causes of vibration and how to verify these causes by testing are discussed.

RAIL

77-2184

Reduction of Railway Noise with Composite Concrete Rails

J. Halpenny

Earth Physics Branch of the Dept. of Energy, Mines and Resources, Ottawa, Ontario, Canada, High-Speed Ground Transp. J., 11 (2), pp 173-175 (Summer 1977) 4 refs

Key Words: Railroad tracks, Noise reduction

Noise due to high speed trains can be greatly reduced by the use of a suitable track structure. A rail with increased stiffness and mass allows the use of much more flexible mountings than are possible with conventional rails. Vibration of the ground and track structure, the most difficult type of sound to handle, is isolated at source. The track will hold a more precise alignment longer, and demands on the foundation are less severe. The technique requires advances in concrete technology, but will make rail systems much quieter.

REACTORS

(Also see No. 2145)

77-2185

Seismic Soil-Structure Interaction Effects at Humboldt Bay Power Plant

J.E. Valera, H.B. Seed, C.F. Tsai, and J. Lysmer
Dames & Moore, San Francisco, CA, ASCE J. Geotech. Engr. Div., 103 (GT10), pp 1143-1161 (1977)
15 figs, 4 tables, 10 refs

Key Words: Nuclear power plants, Earthquake response, Seismic design, Interaction: soil-structure

The results of a study of the distribution of ground motions and structural response in the Humboldt Bay Nuclear Power Plant during the Ferndale earthquake of June 7, 1975 are presented. Based on a knowledge of the motions recorded at the ground surface in the free-field, computations are made to determine the characteristics of the motions likely to develop at the base of the buried reactor caisson at a depth of 85 ft below the ground surface and within the Refueling Building at the ground surface level.

ROAD

(Also see No. 2087)

77-2186

Crash Testing of Experimental Safety Vehicles, Volume II. Renault Basic Research Vehicle

N.J. DeLey
Calspan Corp., Buffalo, NY, Rept. No. CALSPAN-ZP-5857-V-2-Vol-2, COT-HS-802 380, 185 pp (May 1977)
PB-267 966/0GA

Key Words: Collision research (automotive), Crashworthiness, Test data

Results from two crash tests of the Renault Basic Research Vehicle (BRV) are presented. The tests were a left front oblique impact with a rigid 30-degree angled barrier at a speed of 42.5 MPH, and a 75-degree right side impact of the same BRV by the front of a production Renault R-12 automobile at a speed of 31.3 MPH. The objective of the tests was to evaluate the safety performance of the Renault BRV from the vehicle and dummy occupant responses measured in the crashes.

77-2187

Application of Military Vibration Standards to Public Transport Vehicles

G.F. Capponi
ATM Public Transport of Milan, Italy, J. Environ. Sci., 20 (5), pp 25-28 (Sept/Oct 1977) 7 figs

Key Words: Vibration tests, Buses (vehicles), Standards and codes

The objective was to establish a tentative vibration test specification for the ticket machines used on ATM buses. A vibration simulation criterion is described, developed following MIL-STD-810B and considering acceleration measurements made on ATM buses (Public Transport of Milan).

77-2188

An Investigation of Some Responses of an Out-of-Position Driver in an ACRS-Equipped Oldsmobile during Crash Induced Bag Deployment

D.J. Bliss
Office of Vehicle Systems Res., National Highway Traffic Safety Admin., Washington, D.C., Rept. No. DOT-HS-802 315, 69 PP (May 1977)
PB-267 951/2GA

Key Words: Collision research (automotive), Air bags (safety restraint systems), Test data

A study was conducted to investigate the undesirable side effects of inflating a driver air bag system against a forward positioned occupant. The study was at least suggested by an accident which occurred in February 1976 in Memphis, TN, in which the driver of an ACRS-equipped Oldsmobile died as the car struck a utility pole at a speed below the 30 mph design speed of the system. A series of curb rideover tests and a pole impact test were conducted to consider the general problem of occupants positioned forward against inflating air bags and specifically to note any similarities with the Memphis accident.

77-2189

**Crash Testing of Experimental Safety Vehicles.
Volume I. British Leyland Marina Safety Research
Vehicle**

N.J. DeLeys

Calspan Corp., Buffalo, NY, Rept. No. CALSPAN-ZP-5857-V-2-Vol-1, DOT-HS-802 379, 189 pp (May 1977)

PB-267 965/2GA

Key Words: Collision research (automotive), Crashworthiness, Test data

Results from two crash tests of Phase I Marina Safety Research Vehicles (SRV) developed by British Leyland Motor Corp. are presented. The tests were a central head-on collision of a Marina SRV with an AMF experimental safety vehicle at a closing speed of 60 MPH, and a 90-degree side impact of another Marina SRV by a modified production Marina automobile at a speed of 30 MPH. The objective of the tests was to evaluate the safety performance of the Marina SRVs from the vehicle and dummy occupant responses measured in the crashes.

ROTORS

(Also see No. 2118)

77-2190

Finite Element Stability Analysis for Coupled Rotor and Support Systems (Part 3)

K.H. Hohenemser and S.K. Yin

Dept. of Mech. Engrg., Washington Univ., St. Louis, MO, Rept. No. NASA-CR-152024, 47 pp (June 1977)

N77-26079

Key Words: Rotors, Supports, Stability, Finite element technique

The effects of fuselage motions on stability and random response were analytically assessed. The feasibility of adequate perturbation models from non-linear trim conditions was studied by computer and hardware experiments. Rotor wake-blade interactions were assessed by using a 4-bladed rotor model with the capability of progressing and regressing blade pitch excitation (cyclic pitch stirring), by using a 4-bladed rotor model with hub tilt stirring, and by testing rotor models in sinusoidal up to side flow.

77-2191

Effect of Inertia Moment on Critical Speed Calculation of Rotating Shafts (Effetto del Momento Rad-

drizzante sul Calcolo Delle Velocità Critiche di Alberi Rotanti)

B. Atzori

Ist. de Costruzione di Macchine, Bari Univ, Italy, Rept. No. HC A02/MF A01, 12 pp (Oct 16, 1976)

refs

(In Italian)

N77-25544

Key Words: Rotors, Shafts, Critical speed, Inertial forces

The effect of taking into account the lateral inertia in the computation of critical speeds of rotating shafts was analyzed. The power method, Von Borowicz's method, Dunkerley's method, and the matrix displacement and force methods were considered. Some procedures for extending the validity of the examined methods are described after analyzing the mathematical implications due to the presence of negative eigenvalues.

77-2192

The Effect of Nonlinear Internal Damping on the Stability of Simply Loaded Shafts (Zur Stabilität einfacher besetzter Wellen mit nichtlinearer innerer Dämpfung)

P. Hagedorn, H. Kühl, and W. Teschner

Institut für Mechanik, Technische Hochschule Darmstadt, Hochschulstrasse 1, D-6100 Darmstadt, Fed. Rep. of Germany, Ing. Arch., 46 (3), pp 203-212 (1977) 3 refs

(In German)

Key Words: Rotors, Internal damping, Stability

The destabilizing effect of linear internal damping on rotating shafts with a single disc is well-known. Internal damping forces can however in general not be well described by linear functions, but may only be produced with some accuracy with nonlinear terms. In this paper, nonlinear internal damping as well as nonlinear restoring forces are considered, the stability of the vertical and of the horizontal shaft are discussed and non-trivial stationary solutions are also examined. The obtained results confirm to a certain extent the behavior of rotating shaft found by Tondl.

77-2193

A Method for Estimating the Condition that a Rotor Can Pass Through Resonance

K. Matsuura

Hitachi Res. Lab., Hitachi, Ltd., Hitachishi, Japan, Bull. JSME, 20 (145), pp 801-810 (July 1977) 14 figs, 9 refs

Key Words: Rotors, Critical speed

A rotor accelerated across a resonance, which possesses linear properties with a single degree of freedom, excited by an unbalanced rotating mass is considered. It is said that by investigating the non-stationary transitions of motion of a rotor under the critical condition, it can be found whether or not a rotor can pass through resonance or not. It is possible to formulate the condition; and an expression for estimation.

SPACECRAFT

77-2194

Noise Reduction Evaluation of Grids in a Supersonic Air Stream with Application to Space Shuttle

J.M. Seiner, J.C. Manning, P. Nystrom, and S.P. Rao
Langley Res. Ctr., NASA, Langley Station, VA., Rept. No. NASA-TM-X-74034, 36 pp (May 1977) refs N77-25913

Key Words: Spacecraft, Launching, Noise reduction

Near field acoustic measurements were obtained for a model supersonic air jet perturbed by a screen. Noise reduction potential in the vicinity of the space shuttle vehicle during ground launch when the rocket exhaust flow is perturbed by a grid was determined. Both 10 and 12 mesh screens were utilized for this experiment, and each exhibited a noise reduction only at very low frequencies in the near field forward arc.

77-2195

An Evaluation of Reaction Wheel Emitted Vibrations for Space Telescope

Sperry Flight Systems, Phoenix, AZ, Rept. No. NASA-CR-150303; Publ-71-0989-00-00, 108 pp (Mar 1977)
N77-26181

Key Words: Spacecraft components, Vibration measurement

Emitted forces and moments characteristics of the Space Telescope Reaction Wheel Assembly (ST RWA) were measured under room temperature and pressure, thermal extremes, and vibratory conditions. The RWA/Emitting Vibration Measurement Fixture was calibrated statically and dynamically, and background noise was measured with ST RWA not operating. A base line set of forces and moments of the ST RWA along and about three mutually perpendicular axes were recorded at room ambient.

77-2196

Identification of Natural Frequencies and Modal Damping Ratios of Aerospace Structures from Response Data

C.D. Michalopoulos

Dept. of Mech. Engrg., Houston Univ., TX, Rept. No. NASA-CR-151419; TR-NC-1, 36 pp (Nov 1976) N77-26532

Key Word: Spacecraft, Natural frequencies, Modal damping

An analysis of one and multidegree of freedom systems with classical damping is presented. Definition and minimization of error functions for each system are discussed. Systems with classical and nonclassical normal modes are studied, and results for first order perturbation are given. An alternative method of matching power spectral densities is provided, and numerical results are reviewed.

TURBOMACHINERY

(Also see Nos. 2050, 2060, 2065)

77-2197

Supersonic Transport Noise Reduction Technology Program - Phase II, Volume 2

S.B. Kazin, E.J. Stringas, J.T. Blozy, V.L. Doyle, and R.B. Mishler
Aircraft Engine Group, General Electric Co., Cincinnati, OH, Rept. No. R75AEG362-Vol-2, FAA-SS-73-29-2, 470 pp (Sept 1975)
AD-B010 469/5GA

Key Words: Turbomachinery noise, Noise reduction, Supersonic aircraft

Both compressor and turbine noise were studied in the turbomachinery noise reduction areas. A 3-stage low pressure compressor with variable-flap inlet guide vanes was tested at General Electric's outdoor test site. A hybrid inlet, which employs airflow acceleration suppression in combination with wall acoustic treatment, was investigated as the suppression device for all three noise monitoring point operating conditions. The effect of auxiliary inlets on noise leakage and suppression was studied for takeoff mode. Also, variable inlet guide vane flaps were used to reduce area and generate high passage Mach numbers of another means of compressor noise suppression. Turbine noise was studied using a J85 engine with massive inlet suppressor and open nozzle to unmask the turbine. Second-stage turbine blade/nozzle spacing and exhaust acoustic treatment were investigated as means of turbine noise suppression.

ANNUAL AUTHOR INDEX

- A -

Abatan, A.O.	1136	Amies, G.	1736, 1737, 1738, 1739	Au-Yang, M.K.	327, 1214
Abbas, B.A.H.	743, 1292	Anagnostopoulos, S.A.	174, 798	Ayers, W.D.	508
Abdel-Ghaffer, A.M.	356, 1275	Anand, G.V.	982	Ayres, D.J.	281
Abe, T.	1397	Andersen, C.M.	949	Azad, E.	769
Abla, M.A.	1796	Anderson, A.L.	435	Azar, R.C.	2122
Abrahamson, A.L.	2038	Anderson, G.L.	1445, 1763	Azzoni, A.	426
Abrahamson, G.R.	38	Anderson, J.S.	1319, 1779		
Abromavage, J.C.	501	Anderson, M.S.	18		
Adachi, T.	1814	Anderson, W.D.	1808		
Adams, G.H.	1546, 1547	Anderson, W.J.	1447, 1803, 2029	Baade, P.K.	1509
Adams, L.	1024	Ando, Y.	594, 788, 1486	Babu, P.V.T.	1266
Adams, R.D.	106	Andresen, J.A.	1208	Bachschmid, N.	1692
Adeli-Rankoohi, H.	655	Andrews, G.C.	1133	Backaitis, S.	1366
Adi Murthy, N.K.	320	Aneja, I.K.	1363	Backmann, J.N.	793
Adubi, F.A.	2137	Angevine, E.N.	1356, 1357	Badgley, R.H.	222, 224, 409
Agnon, R.	581, 1003, 1004, 2023	Angevine, O.L.	1198	Badlani, M.L.	1406
Agrawal, B.N.	16	Aoki, I.	438	Baghdadi, A.H.A.	2080
Agrawal, F.N.	626	Aoyama, H.	644	Bai, K.J.	279
Ahlbeck, D.R.	1213	Apaydin, T.A.	748	Baig, M.I.	329
Ahmad, A.	1060	Apsel, R.J.	802	Bailey, J.R.	182, 183, 184
Ahmadi, G.	1482, 1538	Arakawa, T.	1390		1531, 1891
Åhrlin, U.	1512	Ardayfio, D.	142, 739, 844	Bainum, P.M.	1956
Aiello, G.F.	1570	Ariaratnam, S.T.	1066	Baird, B.	228
Akers, A.	1454	Arima, K.	1390	Baker, W.E.	470, 496, 1827, 1921
Akerström, T.	1952	Aristizabal-Ochoa, J.D.	1347, 1834	Balaam, E.	1588
Akesson, B.A.	678	Armstrong, F.W.	57	Balachandra, M.B.	1597
Akizuki, K.	1748	Armstrong, J.H.	584	Balachandran, C.G.	860
Akkas, N.	1331	Arnold, P.	171, 173	Baladi, G.Y.	1569
Aksu, G.	338	Arnoldi, R.A.	307	Baldwin, J.L.	636
Albertini, C.	196	Arora, J.S.	12, 836, 1044, 1197	Balke, R.W.	809
Albrecht, D.	1802, 1966	Arulif, C.L.	1844	Ballagh, K.O.	860
Albrecht, D.M.	1955	Arya, S.C.	1685	Balmford, D.E.H.	1365
Alexandridis, A.A.	1873	Aryafar, A.	2111	Balombin, J.R.	1862
Alfredson, R.J.	1687	Ash, J.E.	2047	Banerjee, D.	2175
Ali, R.	338	Ashe, W.A.	507	Banks, D.O.	1294
Al-Jumaily, A.M.	2147	Ashley, H.	555, 745	Bannister, R.H.	305
Allaire, P.F.	306, 886, 1307, 1449	Ashworth, R.P.	935, 1034	Baratono, J.	1233
Allen, R.R.	147, 1132, 1166	Asnani, N.T.	78	Barber, R.B.	136
Alt, R.E.	1410	Atalay, M.B.	983	Barcilon, V.	516
Althof, W.	1600	Atencio, A., Jr.	1994	Bareket, M.	1260
Alwar, R.S.	320	Atzori, B.	2067, 2191	Barger, J.E.	23
Amba-Rao, C.L.	2109	Augustitus, J.A.	2030	Barker, L.K.	1242
Ambati, G.	129	Aurich, H.	1161	Barnard, B.W.	1376

- B -

Barone, M.R.	1985	Bennett, J.A.	2027, 2033	Black, H.F.	411
Barr, A.D.S.	1034	Bennett, R.M.	1728	Blackstock, D.T.	345, 681, 682
Barr, G.W.	472	Bennett, R.O.	1369	Blakney, D.F.	547
Barrett, D.K.	620	Benson, P.R.	780	Blanc, R.H.	711
Barrett, J.R.	639	Benson, R.D.	2172	Blanck, M.W.	1733
Barrett, L.E.	306, 886, 1307, 1454	Bently, D.E.	1087	Blanks, H.S.	889
Barschdorff, D.	1277, 1940	Berendt, R.D.	665	Bleich, H.H.	966, 1828
Bartel, H.D.	856	Berge, B.	299	Blevins, R.D.	20, 942
Bartenwerfer, M.	581, 1003, 2023	Berger, B.S.	775	Bliss, D.B.	548
Bartesch, H.	1433	Berger, E.	2097	Bliss, D.J.	2188
Bartlett, J.C.	638	Bergman, L.A.	479	Blotter, P.T.	136, 1756
Barton, C.K.	1491	Berkof, R.S.	1894	Blouin, S.E.	494
Barton, J.R.	1772	Berman, A.	1879	Blozy, J.T.	2163, 2197
Basas, J.E.	1888	Berman, C.H.	549	Blumenfeld, D.E.	1912
Basavanhally, N.	2139	Bernard, J.E.	1477	Boatright, K.E.	653
Bass, H.E.	1943	Bernard, J.P.	815	Boch, D.C.	1080
Bastow, D.	1506	Bernard, M.C.	790	Bodley, C.S.	226
Bastow, P.	417	Bernhagen, J.R.	368	Bogdanoff, J.L.	1033, 2079
Basu, P.K.	1829	Berry, J.C.	837	Bohm, G.J.	1215
Bates, C.L.	321, 946	Berry, V.L.	1503, 1553	Böhm, R.	2070
Bathe, K.J.	379	Bert, C.W.	766, 1145, 1334	Bohn, M.S.	202, 1674
Bauer, A.B.	933, 1914	Bertero, V.V.	1102, 2008	Bojadziev, G.N.	650
Bauerhop, H.	1601	Bertrand, J.C.	873	Boland, J.S., III.	362
Bauernfeind, V.	1383	Beskos, D.E.	693	Boland, P.	622
Baum, J.H.	2033	Bessey, R.L.	533	Bolding, R.	1840
Baumeister, K.J.	101	Betz, E.	1572	Bolds, P.G.	620
Baxa, D.E.	1591	Beysens, A.	616	Bolen, L.N.	1943
Bayazitoglu, Y.O.	1395	Bezler, P.	322	Bonderson, L.S.	1725
Baylac, G.	93, 94	Bhat, B.R.	1146	Booth, E.T.	182, 1531
Beards, C.F.	2089	Bhat, S.T.	1141	Borza, D.	1976
Becker, J.M.	765, 1369	Bielak, J.	701	Botman, M.	104
Becker, R.J.	344	Bielawa, R.L.	869	Bouts, D.	1443
Beckett, R.E.	2127	Bieniek, M.P.	1430, 1471	Bowes, M.A.	1855, 1879
Beemer, R.L.	501	Biereichel, H.	1931	Bowles, J.V.	1496
Beer, R.	541	Bies, D.A.	66, 1590, 1599, 1623, 1691, 1799, 1950	Bowman, H.F.	583
Beercheck, R.C.	714	Biggs, J.M.	173, 174, 175, 765, 798, 982	Boxwell, D.A.	27, 361
Beeston, H.E.	1267	Bigret, R.	308	Boyce, W.	1221
Bekofske, K.L.	833	Billaud, J.F.	455	Boyd, D.E.	1644
Belek, H.T.	1309	Billingsley, J.	722	Brach, R.M.	1245, 1289
Beliveau, J.G.	980, 1676	Bily, M.	890	Bradshaw, J.C., III.	1551
Bell, J.	732	Birchak, J.R.	1424	Braess, H.H.	834
Bell, J.F.W.	129	Bisconti, N.	383	Bragg, E.E.	288
Belofske, K.L.	1863	Bishop, D.E.	659, 793	Braha, J.	52
Belytschko, T.B.	197, 587, 856, 1126	Bishop, R.E.D.	225, 420, 1291, 2090	Bramwell, A.R.S.	567
Bender, E.K.	1667	Bismarck-Nasr, M.N.	771	Brandt, D.E.	1784
Benham, R.A.	637	Bjorheden, O.	1715	Brandt, K.	1037
Benitou, J.J.	2140	Björkman, M.	1512, 1746	Braun, S.	1774
Bennekers, B.	1498	Bjorno, L.	735	Brausch, J.F.	2158
Bennett, B.E.	232			Breinl, W.	1380
Bennett, D.G.	1568			Breitbach, E.	1413, 1414
				Bremer, H.	633

Bremer, R.C., Jr.	820	Bush, A.R.	1969	Cassanto, J.M.	1720
Bresler, B.	2008	Bush, H.G.	1625	Cassaro, M.A.	518
Brien, M.J.	719	Bushnell, D.	1906	Castle, C.B.	1189
Bright, K.	597	Buth, E.	1023	Celep, Z.	779, 2138
Brignac, W.J.	1787	Button, J.W.	1023	Cecil, D.J.	671, 672
Brito, J.D.	877	Butzel, L.M.	671, 791	Cermak, J.E.	800
Britt, J.R.	699	Buxbaum, O.	2007	Chace, M.A.	1257, 1395, 2068, 2069
Broadbent, E.G.	1997	Bycroft, G.N.	1747, 2011	Chadwick, P.	42
Brommundt, E.	427, 2050	Byers, J.F.	1576	Challis, L.A.	1624, 1702
Broner, N.	1687	Byrne, K.P.	1697, 2161	Chalupnik, J.D.	1440
Bronowicki, A.	375, 675, 676, 677	Byrne, R.	1208	Chamis, C.C.	1329, 1810
Brooke, R.N.	31			Champomier, F.P.	711
Brooks, J.J.	482			Chander, S.	1022
Brown, B.E.	459			Chandiramani, K.L.	1973
Brown, D.	879			Chandra, J.	2144
Brown, D.	1007	Cagliostro, D.J.	588	Chandra, R.	112
Brown, D.L.	1372, 2100	Calahan, D.A.	2068, 2069	Chandran, K.B.	128, 363
Brown, F.T.	854	Caldwell, D.W.	478	Chandrasekhar, P.	1607
Brown, G.L.	1705	Caldwell-Johnson, W.H.	1643	Chang, C.J.	853, 1428
Brown, J.M.	755	Calistrat, M.M.	1322	Chang, D.C.	2028
Brown, P.J.	1082	Calkins, D.E.	1105	Chang, E.H.	2025
Brown, R.E.	447	Callegari, A.J.	1121, 1622	Chang, N.	1
Brown, S.M.	1425	Calzado, A.J.	1838	Chang, Y.M.	956
Brown, T.J.	25	Campbell, G.M.	1755	Chang, Y.R.	531
Browne, R.C.	1581	Campbell, K.L.	1368	Chapkis, R.L.	933
Brueck, D.M.	2177	Campbell, J.M.	671	Chapman, P.C.	848
Bruel, P.V.	1689	Campomanes, N.V.	989	Chappell, M.S.	717
Brugh, R.L.	1644	Canavin, J.R.	439	Charity, I.A.	1564, 1641
Brussalis, W.G.	829	Candel, S.M.	1841	Charoenree, S.	79
Bryan, M.E.	835	Cannarozzi, A.A.	1050	Chatopadhyay, S.	113
Bryden, J.E.	1684	Cansdale, R.	1704	Chaudhuri, S.K.	1845, 2110
Bukoveczky, J.	890	Caplan, W.F.	434	Chavez, H.R.	1349
Bull, H.L.	715	Capponi, G.F.	2187	Chea, W.	1176
Bull, M.K.	1640	Capps, D.S.	1758	Cheilas, N.	1222
Bullard, O.J.	402	Capranica, R.R.	65	Chelapati, C.V.	1614
Bultzo, C.	892	Capriz, G.	433	Chen, C.K.	168, 799, 1917
Bundorf, R.T.	601	Caputo, M.	119	Chen, E.P.	1073
Buono, D.F.	219, 1206	Caravani, P.	1246	Chen, F.	2052
Burcham, F.W., Jr.	1999	Card, M.F.	1625	Chen, F.C.	125
Burchill, R.F.	899	Cardia, S.	63	Chen, F.Y.	531, 1055, 2129, 2130
Burdess, J.S.	284	Carlson, D.R.	1462	Chen, H.	1409
Burmeister, L.	181	Carmichael, A.J.	1572	Chen, J.C.	456
Burney, F.A.	185, 2182	Carmichael, D.	1890	Chen, L.H.	1420, 1421, 1895, 1899
Burns, C.D.	359	Carne, T.G.	2033	Chen, P.J.	265
Burns, E.M.	26	Carpenter, A.B.	1009	Chen, R.P.	790
Burrin, R.H.	1167, 1168	Carr, R.W.	209	Chen, S.S.	532, 941, 1313, 1813, 1965, 1982
Burroughs, C.B.	2146	Carta, F.O.	307	Chen, T.L.C.	1334
Burrows, C.R.	947, 1970	Caruso, H.	503, 580	Chen, Y.H.	576
Burton, R.T.	1534	Caruthers, J.E.	816, 1113		
Burton, T.E.	20	Casandjian, G.	62		
Burwell, G.R.	509	Caspi, A.	146		

- C -

- D -

Chen, Y.N. 93, 94, 1158
Cheng, F.Y. 1359
Cheng, S. 747
Cheng, W.H. 1099, 1238
Cheng, Y.F. 1760
Cherchas, D.B. 1602
Cherng, J.G. 544
Chestnutt, D. 1461
Cheung, Y.K. 1346
Chi, C.C. 1381
Chi, F.H. 150
Chia, C.Y. 1820
Chiang, T. 224
Chiapetta, R.L. 856
Chien, C.F. 241
Childs, D. 1451
Childs, D.W. 218
Chisholm, R. 712
Chon, C.-S. 2096
Chonan, S. 81, 83, 777
Chopra, A.K. 702, 796,
 1677, 1678
Chopra, P.S. 589
Chorkey, W.J. 1348
Chou, C.C. 851
Chou, P.C. 95, 297, 1604, 1809
Chretien, J.P. 615
Christian, J.T. 492
Christiansen, H.N. 459
Christiansen, V.T. 1756
Christmann, C. 2015
Chrostowski, J.D. 1180
Chu, K.H. 357
Chu, S.C. 2127
Chun, K.S. 549
Chung, H. 1813, 2145
Chung, J.Y. 1386
Chung, T.J. 325
Clamen, M. 2131
Clapis, A. 426
Clark, N.H. 1596
Clark, R.N. 1225
Clarke, J.D. 337
Clevenson, S.A. 1190
Cochery, P. 481
Cockerham, G. 1047
Cohen, H. 1303
Cohen, M.J. 11
Cohen, R. 190
Cole, E. 1754
Collacott, R.A. 49, 999
Collins, H.D. 738
Collyer, M.R. 558
Confer, V.J. 252
Connelly, W.H. 1214
Connor, J.G., Jr. 1502
Cooper, W.D. 1891
CooperRider, N.K. 1209
Coppendale, J. 106
Corley, D.M. 840
Corliss, E.L.R. 665
Cornell, R.W. 522
Cornillon, C. 1061
Corotis, R.B. 1752
Corr, R.B. 649
Corradi, L. 8
Costantino, C.J. 703
Costello, G.A. 1304, 1344
Couchman, J. 732
Coull, A. 1103
Courtine, D. 710
Cowan, S.J. 549
Cox, P.A. 496
Cox, W. 1776
Cozzarelli, F.A. 710
Craggs, A. 451
Craig, A. 50
Craig, R.R., Jr. 853, 1428
Crampton, F.J.P. 445
Crandall, S.H. 1187
Crews, S.T. 2118
Crocker, M.J. 969
Cromer, J.C. 641
Cronkhite, J.D. 1503, 1553, 1555
Croome, D.J. 817
Crossley, F.R.E. 2122
Crouch, R.W. 2003
Crowe, C.T. 891
Culver, C. 1681
Cummings, A. 758, 1318
Cunniff, P.F. 996
Cunningham, H.J. 1134, 1554
Cunningham, J. 1614
Cunningham, R.E. 90, 922
Cunny, R.W. 523
Curtiss, H.C., Jr. 559, 973
Cusano, C. 887
Cushing, W.M. 26
Czarnecki, R.M. 168, 799
Dahl, P.R. 692
Dalal, J.S. 1766
Danckert, H. 394
Daniel, B.R. 227
Daniel, W.J.T. 1721
Danisch, R. 824
Darden, C.M. 1495
Das, A. 624
Das, Y.C. 1272
Dasa, N. 646
Dasgupta, G. 1276
Da Silva, M.R.M.C. 1710
Dat, R. 730, 806
Datta, P.K. 764
Datta, S. 120
Davern, W.A. 1481
Davidson, J.K. 1529
Davidson, J.W. 2048
Davies, H.G. 878
Davies, J.M. 1950
Davies, P.B. 920
Davies, W.G.R. 445
Davis, P.J. 377
Davis, P.K. 885
Davis, W.S. 524
Davy, B.A. 843
Dawson, B. 896
Day, F.D. 848
De, S. 483
Dean, D. 1167
Dean, P.D. 526, 1122
DeCapua, N.J. 466
Degen, P. 195
Degener, M. 1412
De Hoog, F.R. 235
DeJong, R. 144
Delaney, M.E. 464
DeLeys, N.J. 1019, 1020, 1388,
 1389, 2186, 2189
Della Pietra, L. 1474
Deloach, R. 167, 564
Delph, T.J. 273
DelValle, R.J. 1892
Demchak, L. 1029
Dempsey, T.K. 1190
Dennett, R.H. 613
Derby, T.F. 573, 1058
DesForges, D.T. 1876
De Silva, C.W. 85

Desjardins, R.A.	804	Dowson, D.	.924, 925	Eisley, J.G.	.77		
Desmarais, R.N.	1728	Doyle, G.R.	1212	El Baradie, M.A.	186		
Devers, D.A.	226	Doyle, V.L.	2163, 2197	Elishakoff, I.	1142		
DeVries, M.F.	810	Dragsten, P.R.	.65	Ellen, C.H.	1764		
Diana, G.	1621	Drakatos, P.A.	.489, 490	Ellingson, E.F.	367		
Dib, G.M.	334	Drake, J.L.	.699	Elliott, J.L.	2126		
DiBlasi, A.	791	Drane, D.A.	.1671	Ellis, J.R.	1478, 1526		
Dickerson, J.	1048	Dransfield, P.	.1376, 1700, 1701	Ellis, R.	2024		
Diehl, G.M.	1092	Drechsler, J.	.410	Ellis, R.W.	2178		
Diercks, A.D.	1941	Drenick, R.F.	.1860	Ellyin, F.	1607		
Dieterich, D.A.	2034	Drewyer, R.P.	.1685	Elmasri, M.Z.	132		
Dietman, H.	1578	Dubey, R.K.	.330	Elson, J.P.	190		
Dietrich, R.	436	Dubowsky, S.	.1131, 1132	Emerson, P.D.	.182, 184, 1531, 1891		
DiGiorgio, A.	63	Dufort, R.H.	.365	Emery, A.F.	122		
Dilger, W.	132	Duggin, B.W.	.506	Emery, B.	1780		
DiMaggio, F.L.	1828	Dugundji, J.	.984	Emmerling, J.J.	.833, 1385		
DiMasi, F.P.	.867, 868	Duke, K.	.1701	Endo, M.	.865, 866		
Dimmick, B.W.	1363	Dukes, R.E.	.1202	Engblom, J.J.	.695		
Dini, D.	63, 382	Dumanoglu, A.A.	.645	Engel, P.A.	.1571		
Dissen, H.	199	Dunens, E.K.	.1887	Engels, R.C.	.1065		
Dittmar, J.H.	1002, 2021	Dunet, G.	.721	Enserink, E.	.1366		
Dittrich, W.	155	Dung, L.	.517	Eriksson, L.J.	.390		
Djiauw, L.K.	1927	Dungar, R.	.1983	Ernsberger, K.	.632		
Dobbs, N.	2066	Dunn, D.G.	.671, 672, 791	Erskine, J.B.	.404		
Dobbs, S.K.	1844	Dunn, W.H.	.1903	Ertepinar, A.	.2143		
Dobrzynski, W.M.	2002	DuPont, J.F.	.1384	Ervin, R.D.	.1219, 1220, 1477		
Dodd, V.R.	900	Durelli, A.J.	.906, 909	Esche, D.	.818		
Dodds, C.J.	453	Durham, D.J.	.668	Eshel, R.	.977		
Doggett, R.V., Jr.	.554, 1499, 1554, 1793	Durocher, L.L.	.111	Esparza, E.D.	.470, 1921		
Dokumaci, E.	1880	Dusel, J.P.	.1223	Essary, J.D.	.1442, 1942		
Doll, W.	282	Dykstra, R.A.	.1591	Etsion, I.	.2115		
Doll, R.W.	519	Dym, C.L.	.166, 709, 1688	Evans, K.E.	.684		
Dolumaci, E.	1606	Dzygadlo, Z.	.151, 1150, 2020	Evans, K.W.	.1462		
Doman, G.S.	1686	- E -		Evensen, D.A.	.737		
Donato, R.J.	34	Eade, P.W.	.1514	Evensen, H.A.	.1662		
Done, G.T.S.	.976, 1188	Earles, S.W.E.	.1632	Everett, W.D.	.498		
Donea, J.	138	Eberhardt, A.C.	.195, 1398	Eversman, W.	.98		
Donovan, N.C.	1853	Eby, T.L.	.619	Eversole, K.B.	.1243		
Doolan, P.	.185, 188	Eckert, W.L.	.182, 1531	Everstine, G.C.	.1557		
Dooley, L.W.	1310	Eckmann, J.P.	.13	Ewing, D.K.	.1938		
Dorien-Brown, B.	1722	Eden, D.	.1682	Ewins, D.J.	.310		
Dostal, M.	1288	Edgington, F.M.	.499	- F -			
Dougan, A.C.	1427	Edwards, C.L.	.91	Fachbach, H.A.	.203, 1216		
Dougherty, M.R.	1680	Edwards, D.C.	.91	Fagel, L.W.	.1680		
Doughty, S.	1039, 1226	Edwards, J.C.	.1460	Fagerlund, A.C.	.324		
Douglas, B.E.	495	Edwards, T.	.1784	Fahy, F.J.	.781, 1824, 1825		
Dowell, E.H.	.107, 875, 1293, 1832	Eidson, R.L.	.325	Falade, A.	.2085		
Downham, E.	406	Eisenberg, I.M.	.1918				
Downs, B.	744						

Falco, M.	1621	Foley, W.M.	59	Fujiwara, N.	997, 1192
Fandrich, R.T., Jr.	514	Föller, D.	1663	Fujiwara, Y.	1396
Farassat, F.	25, 1995	Fong, A.	1269	Fukano, T.	1203, 1204
Farrell, J.J.	642	Fong, S.K.L.	1101	Fukuda, H.	37
Farshad, M.	1963	Foppe, G.F.	1786	Fukuoka, H.	47, 272
Faulkner, L.L.	927	Ford, C.A.	2039	Funaro, J.	1471
Faulkner, M.G.	1611	Ford, M.B.	264	Funk, P.E.	887
Fawcett, J.N.	1465	Foreman, D.A.	556	Furman, J.E., Jr.	1225
Fawzy, I.	420, 1271	Fornallaz, P.	1284	Furrer, H.	195
Feaster, L.	991	Forrai, L.	421		
Feger, D.	1269	Fortescue, P.W.	1610		
Fehl, C.	1720	Forzono, C.J.	1868		- G -
Feiler, C.E.	1461	Foss, R.N.	543		
Feix, M.	667	Foster, J.E.	1165	Gabri, B.S.	1934
Felske, A.	1281	Foughner, J.T., Jr.	1791	Gade, R.H.	2168
Felton, L.P.	2048	Foutch, D.A.	1358	Gaffey, T.M.	809
Feng, T.T.	12, 1044	Fowler, J.R.	1876	Gaffney, J.	1466
Feng, W.W.	937	Fox, E.N.	918	Galatisis, A.G.	441
Ferer, K.M.	930	Fox, G.L.	530	Gale, J.G.	300
Ferrante, J.G.	629	Fox, R.L.	603	Galka, A.	1326
Ferrante, M.	634	FraeijnsdeVeubeke, B.	627	Galloway, W.J.	659, 660, 2000
Ferre, M.	1783	Fralich, R.W.	1228	Gamon, M.A.	1172
Ferritto, J.M.	471	Francois, P.	2108	Gangwani, S.T.	807
Fertis, D.G.	1457	Frank, R.A.	173, 174, 765	Gaonkar, G.H.	992, 1709
Fields, J.M.	1517	Franke, M.E.	943	Garba, J.A.	456
Filetti, E.G.	2101	Franz, L.	1161	Gardner, T.N.	1131
Filipich, C.	1651	Franzmeyer, F.K.	161	Garg, S.C.	1717
Filippov, A.P.	1446	Farey, J.L.	73, 899	Garg, V.K.	2025
Filler, L.	791	Fraser, R.	1089	Gargiulo, E.P., Jr.	1108
Finch, R.D.	1436	Fraser, W.H.	1969	Gariboldi, R.	426
Findlay, A.	1938	Freeman, C.E.	2164	Garner, H.C.	1669
Fink, M.R.	1992	Freeman, D.	511	Garrellick, J.M.	1343
Finkelstein, W.	1378	Freeman, S.A.	799, 1833	Garrett, R.M.	2078
Finley, T.D.	564	Frei, O.	2035	Garrison, D.W.	1755
Fiore, N.F.	69	Fresa, F.	2067	Gasch, R.	431
Firth, D.	2062	Fricke, F.R.	1487	Gasparett, M.	1621
Fisher, M.J.	1262	Fricke, J.	1620	Gasparini, D.	172
Fisher, W.E.	1679	Friedmann, P.	378, 1114, 1730,	Gast, Th.	1443
Fistedis, S.H.	591		1959, 2173	Gates, N.C.	2082
Fitremann, J.	56	Friedrich, R.	1277	Gatley, W.S.	1893
Fitzpatrick, J.A.	2134	Frith, R.H.	1613	Gatto, M.	1302
Flanagan, P.F.	670, 1561	Fritz, J.T.D.	870	Gaub, F.	1224
Flanders, D.C.	390	Frohrib, D.A.	844	Gaukroger, D.R.	1670, 1704
Flandro, G.A.	1124	Frohrib, F.A.	201	Gauaurd, G.	1915
Fleeter, S.	1113	Frölich, P.	1549	Gauvin, R.	2123
Fleming, D.P.	90, 222, 922	Froseth, S.E.	1063	Gayed, Y.K.	2080
Fletcher, N.H.	257	Frutschi, H.U.	1384	Gazetas, G.	171, 173, 686
Fleury, W.M.	484	Fuehrer, R.R.	1626	Gebben, V.D.	1735
Flis, W.J.	297, 1604	Fujimoto, Y.	386	Gehrig, J.	1284
Flower, J.O.	384, 385	Fujimura, Y.	1800	Geissler, H.	2031
Flower, J.W.	1314	Fujiwara, K.	1486	Gelman, A.P.	2148

Gelos, R.	1444	Greene, B.	1738, 1739	Hadden, W.J., Jr.	970
Genin, J.	859	Greene, J.E.	1387	Hadjian, A.H.	705, 708
George, P.J.	1340	Gregorian, V.	1373	Hagan, T.N.	1567
Gersbach, V.S.	1367	Gregory, R.A.	554, 2167	Hagedorn, P.	2192
Gersch, W.	657, 1750	Greif, R.	36, 341	Hahn, E.J.	1616
Getline, G.L.	561	Greiner, H.	2124	Haidl, G.	2162
Ghali, A.	132	Greitzer, E.M.	1459	Hain, K.	1550
Ghazzaly, O.I.	80	Griffin, M.J.	569, 570, 995,	Haines, D.W.	1
Giardino, D.A.	504		1345, 1504	Haisler, W.E.	852
Gibbons, R.T.	1574, 1698	Griffin, O.M.	901	Hale, A.L.	178, 1038
Gibbs, A.	511	Grinev, V.B.	1446	Hall, F.L.	1745
Gibbs, B.M.	782	Groh, A.R.	1835	Hall, J.R., Jr.	590, 704
Gibs, J.	794	Grooms, D.W.	1548, 1905	Hall, M.	234
Gibson, J.S.	547	Grootenhuis, P.	1519	Hall, W.E., Jr.	1732
Gibson, R.F.	1074, 1075, 2107	Gross, H.	1627	Halleux, J.P.	138
Giers, A.	413	Grossman, D.T.	1786	Halliwell, D.G.	309
Gikadi, T.	581, 2023	Grover, G.K.	2141	Hallman, P.J.	1664
Gilbert, D.	1263	Gruenewald, B.	154	Hallquist, J.O.	937
Gilford, C.L.S.	782	Gubser, J.L.	621	Halpenny, J.	2184
Giuliani, S.	138	Gudat, H.	1939	Hamada, H.S.	1101
Glaser, F.W.	220	Guedes Soares, C.A.P.	915	Hamati, R.E.	2169
Glasgow, D.A.	422	Guendelman-Israel, R.	2170	Hamel, P.G.	152
Glegg, S.A.L.	1262	Guenther, D.A.	927	Hamilton, C.W.	708
Glenn, A.J.	447	Gunter, E.J.	306, 886, 1307,	Hamilton, W.S.	1170
Glick, J.M.	1496		1449, 1454	Hamma, G.A.	502, 725
Goedel, H.	2071	Gunzburger, M.D.	19, 239, 465	Hammond, C.E.	1730, 1793
Goldberg, J.	2135	Gupta, A.K.	749, 1311	Hannebrink, D.N.	1211
Goldelius, R.	720	Gupta, B.P.	931	Hannibal, A.J.	479, 529, 656, 857
Goldman, H.I.	568	Gupta, K.K.	429, 612, 679	Hansen, C.H.	66, 1590
Goldsmith, W.	979, 1642	Gupta, N.K.	1732	Hansen, R.J.	919
Golinski, J.A.	1191	Gupta, P.K.	1448	Happe, A.	1281
Gomperts, M.C.	1342	Gupta, R.K.	1477	Hara, F.	1467
Gongloff, H.R.	631	Gusakov, I.	605	Harari, A.	127
Gordon, H.S.	1872	Gustafson, W.C.	574	Harcrow, H.	1029
Gordon, P.	1609	Guthrie, K.M.	44, 685	Hardin, J.C.	550, 1492
Gordon, P.F.	1923	Gutierrez, J.A.	702	Hardy, A.E.J.	1514
Gorman, D.J.	957, 1153, 1977	Gutierrez, R.	539	Harker, R.G.	894, 2103
Gorman, G.F., III	1832	Gutowski, T.G.	709	Harland, D.G.	464
Gorshkov, A.G.	1143	Guy, T.B.	1660	Harmon, R.P.	902
Gosele, K.	787	Guzman, R.A.	596	Harold, P.F.	2019
Gottlieb, G.	1069			Harper, C.R.	1440
Gould, P.L.	1139, 1829			Harper-Bourne, M.	1262
Goyer, H.G.D.	2017			Harris, J.D.	371
Grab, H.	1374			Harrison, H.D.	1213
Grabec, I.	1200	Habeck, R.	1380	Hart, E.D.	1186
Grabowski, B.	251	Haber, S.	1848	Hart, F.D.	183, 789
Grabowski, S.E.	582	Habercom, G.E., Jr.	663, 664	Hart, G.C.	375, 609, 2048
Graham, S.L.	401	Hadan, G.	977	Hartmann, P.W.	2025
Grashof, M.	876	Haddad, S.D.	1610	Hartz, B.J.	1225
Gray, D.C.	1247	Haddara, M.R.	2080	Hashin, Z.	283, 1768
Greathead, S.H.	417	Hadden, J.A.	1210	Haslinger, K.H.	2135

Hassab, J.C.	233	Hibner, D.H.	219, 1206	Holzweissig, F.	1907, 2015
Hassan, Y.E.	1608	Hickling, R.	213	Homans, B.L.	32
Hasselman, T.K.	275, 375, 609, 675, 676, 677, 1180	Hidaka, T.	103, 1127	Hood, R.A.	464
Hassig, H.J.	855	Higgins, D.S.J.	1195	Hooker, R.J.	1573, 1613, 1661, 1721
Hastings, E.C., Jr.	24	Higgins, T.H.	643	Hooper, W.E.	804
Haug, E.J., Jr.	12, 1044, 1197	Hilber, H.M.	249, 1417, 1882, 2046	Horlock, J.H.	1459
Häusler, N.	1630	Hiller, W.J.	1319	Horvath, M.	207
Haviland, R.W.	175, 982	Hilliard, J.K.	1984	Hosp, E.	2016
Havron, M.D.	2087	Hillquist, R.K.	662	Hou, S.	194
Hawks, R.J.	377	Hino, M.	280	Houbolt, J.C.	1795
Hawthorne, K.L.	2025	Hinton, E.	780	Houghton, J.R.	256, 1081
Hay, J.H.	792	Hirai, H.	1775	Hovanesian, J.D.	906, 909
Hayashi, T.	1800	Hirano, Y.	117, 1070, 1801, 2112	Howard, G.E.	827
Hayden, R.E.	548	Hirao, M.	1886	Howe, M.S.	1816
Hayduk, R.J.	1149	Hirji, F.K.I.	880	Howell, J.F.	1324
Hayek, S.I.	1650, 1909, 2074	Hirschwehr, E.	871	Howell, L.J.	2030
Hayes, M.	689	Hitchings, D.	2121	Howells, R.W.	1556
Hazell, A.F.	1997	Hizume, A.	71	Hribar, A.E.	1064
Hazell, C.R.	1151	Ho, C.H.	77	Hsiao, M.H.	654, 836, 1197
Healy, M.J.	192	Ho, L.T.	1962	Hsu, C.S.	1099, 1238
Heard, W.L.	18	Hobbs, A.E.W.	1106	Hsu, S.T.	110
Heckel, K.	1657	Hobbs, G.K.	482, 1790	Huag, E.J., Jr.	836
Heckl, M.	1563	Hoberock, L.L.	1857	Huang, C.C.	84, 772, 1605
Hedrick, J.K.	1211	Hodder, B.K.	1377	Huang, T.C.	84, 125, 624
Heebink, T.B.	133	Hodges, D.H.	92, 1293	Hubbard, H.H.	863
Hegdahl, T.	266, 267, 268, 269, 270	Hodgetts, D.	414	Hübner, G.	1093
Hegemier, G.A.	1104	Hodgson, D.C.	187	Hud, G.C.	255
Heggie, R.S.	1584	Hodgson, T.H.	1199	Hudachek, R.J.	900
Heidebrecht, A.C.	1157	Hoelscher, H.	158	Hughes, A.D.	976
Heinig, K.	162	Hoffman, J.A.	1579	Hughes, P.C.	1718
Heller, H.H.	2002	Hoffmann, D.	46, 1006	Hughes, T.J.R.	6, 249, 450, 1882
Heller, R.A.	783	Hoffmann, G.	2006	Hugus, G.D.	286
Hermings, R.C.	304	Hoffmann, R.	153	Hull, M.L.	647
Henderson, H.R.	1836	Hogan, B.J.	985	Hull, R.	1209
Henderson, R.E.	1459	Hohenemser, K.H.	2118, 2175, 2190	Hullender, D.A.	401, 1013
Henghold, W.M.	88	Hoitsma, K.L.	2042	Humar, J.L.	1179
Hennessy, K.W.	18	Hokanson, J.C.	475	Hundal, M.S.	343, 1162, 1163
Henry, T.A.	405	Holdsworth, T.M.	729	Hung, Y.Y.	906, 909
Hensle, W.	1277, 1940	Holliday, B.G.	167, 564	Hunter, D.	2093
Herbert, R.G.	2128	Hollin, K.A.	1259	Hunter, T.O.	472
Hermayer, J.	1175	Hollingsworth, L.W.	921	Huntley, I.	1274
Hernalsteen, P.	381	Holmer, C.I.	1094	Hure, D.	768
Herrmann, G.	232, 273, 1327	Holmes, H.K.	167, 564	Hurley, S.R.	1788
Hersh, A.S.	1317	Holmes, P.J.	1418, 2133	Huseyin, K.	432
Hershey, R.L.	643, 2154	Holmes, R.	1288	Hussaini, M.Y.	2109
Hessler, G.F.	1837	Holsapple, D.E.	1785	Huston, R.L.	754, 978
Hetman, M.G.	466	Holton, R.F.	1463	Hutchinson, J.R.	2140
Heusmann, H.	632	Holtz, M.	1967	Hutton, G.B.	1671
Heymann, R.W.	515	Holzer, S.M.	1551, 1740	Hutton, S.G.	1346
				Huttsell, L.J.	1785

Hwang, Y.F.	2039	Jennings, W.P.	1794	Kaestle, H.J.	825
Hwong, S.T.	80	Jensen, F.R.	630, 1719	Kaiser, J.E.	2119
		Jensen, J.J.	1422, 1423, 1896, 1898	Kajimura, Y.	822
		Jensen, J.W.	1488	Kailand, A.	212
- I -		Jensen, P.S.	14	Kalinowski, A.J.	576, 1559
Ibanez, P.	726, 827	Jha, S.K.	215, 1222	Kaliski, S.	277
Ibrahim, R.A.	43	Jido, J.	41	Kamal, M.M.	2030
Ibrahim, S.R.	454	Jobsis, A.C.	1853	Kamat, M.P.	952, 1100
Ichikawa, A.	1805	Jogi, P.N.	40	Kamel, H.A.	1741
Ikui, T.	1285	Johannes, J.D.	1193, 1194	Kamil, H.	263
Ilie, L.	801	Johnson, A.F.	294	Kamperman, G.W.	903
Illingworth, R.	1106	Johnson, C.M.	1032	Kan, C.L.	796, 1178, 1677, 1678
Imaizumi, T.	1401	Johnson, D.A.	1411	Kana, D.D.	1878
Imes, R.S.	2167	Johnson, E.	1333	Kangasabay, S.	1543
Inasaki, I.	1508	Johnson, E.H.	745	Kanarachos, A.	1536
Infante, E.F.	751	Johnson, G.E.	1484	Kane, T.R.	1404
Ingenito, F.	45	Johnson, G.R.	250	Kanematsu, H.	881, 1196
Ingram, J.N.	1095	Johnson, H.W.	830	Kanetaka, S.	1523
Isaacson, D.	3	Johnson, M.K.	1787	Kannel, J.W.	1282
Ishibashi, I.	493	Johnson, W.	993, 1668, 2117	Kaper, B.	243
Ishida, K.	1523	Johnsson, C.A.	1715	Kaplan, B.Z.	2105
Ishida, Y.	1712	Johnston, G.W.	717	Kapur, A.D.	912
Ishihara, A.	1456	Johnstone, N.J.	1464	Karassik, I.J.	1969
Ishioka, K.	1127	Jones, A.D.	1705	Karchmer, A.	1842
Issler, I.	1578	Jones, C.T.	513	Karle, A.P.	1349
Ito, A.	1473	Jones, D.S.	683	Karnopp, D.	147
Ivey, E.S.	260, 2156	Jones, M.H.	1259	Kasemset, C.	1346
Iwan, W.D.	339, 1727	Jones, N.	130, 847, 915, 916, 1954	Kassimali, A.	763, 1125
Iwashige, H.	1759	Jones, P.E.	1489	Kato, K.	788
Iwata, Y.	1164	Jones, P.J.	485	Katsikadelis, J.T.	742
Iwatsubo, T.	408	Jones, R.	808, 1154	Katz, H.	1786
- J -		Jones, R.E.	134	Kaufman, L.	1924
Jackson, C.	423	Jones, R.S.	100	Kaul, M.K.	1415
Jackson, J.D.	1602	Jones, W.N.	609, 621	Kaul, R.K.	273
Jacobs, L.D.	791	Jordan, F.D.	1306	Kausel, E.	492, 1852
Jacquot, R.G.	245, 968, 1165	Joyner, R.G.	602	Kawaguchi, O.	594
Jaffe, L.D.	2043	Judd, S.H.	1699	Kawakami, N.	1407
Jakel, S.	1224	Jung, J.P.	615	Kawakatsu, T.	822
James, D.W.	1780	Jungclaus, D.	1382	Kawatani, M.	795
James, P.K.	1956	Junghans, R.	347	Kayser, K.W.	1033, 2079
Janardan, B.A.	227	Jungowski, M.W.	1319	Kazin, S.B.	145, 1385, 2163, 2197
Jeanmonod, R.	1384	- K -		Keane, A.	1061
Jeffery, R.W.	1997	Kabir, A.F.	1639	Keegan, W.B.	631
Jemielewski, J.	195	Kacena, W.J.	485	Keinholz, D.A.	1405
Jendrzejczyk, J.A.	2145	Kadikar, A.	1024	Keire, H.	716
Jennings, A.	649	Kadlec, J.	586	Keith, R.H.	1355
Jennings, P.C.	596			Kellenberger, W.	718

Keltie, R.F.	1115	Kluwick, A.	242	Kross, D.A.	640
Kempner, J.	1826	Knauer, C.D.	1875	Ku, A.B.	1881
Kenchington, H.S.	495	Knickerbocker, J.L.	2019	Kuak, Y.C.	244
Kennedy, B.J.	1567	Knight, A.L.	1586	Kuczynski, G.C.	69
Kennedy, J.M.	587	Kniskern, J.	727	Kuhar, E.J., Jr.	611, 2072
Kennedy, J.S.	1611	Knott, P.R.	2158	Kühl, H.	2192
Kennedy, R.P.	706	Ko, S.H.	1962	Kuhn, G.F.	109
Kennedy, W.	1977	Ko, W.L.	1827	Kuhn, M.	1354
Kennedy, W.C.	1468	Kobayashi, A.S.	122	Kühner, D.	1949
Kentzer, C.P.	463	Koch, W.	934, 1817, 2120	Kuipers, G.	832
Kerle, H.	1628	Kodama, Y.	1203, 1204	Kukkola, T.	380
Kernevez, J.P.	455	Koenig, R.J.	2001	Kulesz, J.J.	533
Kerr, A.D.	911	Koerner, W.	1182	Kulin, S.A.	1924
Kester, J.D.	975	Kohler, H.	1731	Kulisiewicz, M.	658
Khanna, S.M.	164	Kohli, D.	2093	Kulla, P.	623
Khorzad, N.	91	Kojima, E.	944, 945	Kumar, R.	2144
Khu, K.T.	2091	Koltzsch, P.	347	Kumar, S.	1524
Kiefer, F.W.	1756	Komaroff, N.	1726	Kumar, V.	1111
Kiefer, J.E.	1912	Komatsu, K.	1822	Kuratani, K.	438
Kierkowski, J.	1150	Komatsu, S.	795	Kurowski, G.J.	1294
Kiessling, F.	68	Kondo, S.	594	Kurtenbach, F.J.	1999
Killgoar, P.C.	1927	Koopmann, G.H.	1988	Kurth, U.	1657
Kilmer, R.D.	840	Korecki, T.	438	Kurz, K.	207
Kim, Y.K.	989	Kornecki, A.	107	Kurzweil, L.G.	1520
King, A.C.Y.	1917	Kortum, W.	1908	Kusenberger, F.N.	1772
King, W.F., III	291, 908	Kosloff, D.	262	Kuttruff, H.	131
Kingman, B.C.	1368	Koss, L.L.	1511, 1693	Kuyper, D.J.	482
Kingsbury, H.B.	114	Kossover, D.	2066	Kvaternik, R.G.	1353
Kinney, W.A.	261	Kot, C.A.	2150		
Kinns, R.	722	Kothawala, K.S.	1888		
Kircher, C.A.	1789, 1849	Kounadis, A.	742		- L -
Kirchoff, R.	991	Koutsoyannis, S.P.	1119		
Kirk, C.L.	954	Kouwen, N.	2137	Labes, M.	824
Kirk, R.G.	428, 1206	Koval, L.R.	351	Laenen, E.G.	1955
Kirkhope, J.	96, 929, 1156	Kozlowski, H.	1493	Laithier, B.E.	295
Kirlan, P.	1708	Kozyra, T.W.	1076	Lak, S.	1507
Kirshenboim, J.	1762	Krachman, H.E.	434	Lakin, W.D.	752
Kirsten, P.W.	2005	Kraft, R.E.	101, 761, 1120, 1320	Lakis, A.A.	1646
Kisliakov, S.D.	335	Kraigie, L.G.	178	Lakshminikanthan, R.	1096
Kissenpfennig, J.F.	590, 704	Krajcinovic, D.	2063	Lakshminarayana, B.	2075
Klahs, J.W.	2034	Kramer, E.	430	Lal, S.	2141
Klein, L.	1974	Kramer, J.H.	936	Lalanne, C.	467, 468
Klein, M.J.	1126	Krause, N.	1642	Lalanne, M.	598, 641, 756
Kleinstein, G.G.	239, 465	Krieger, W.	2079	Lambert, R.F.	1063, 1887
Klement, H.D.	430	Krile, T.F.	960	Lambert, R.G.	487
Klepzig, W.	1806	Krings, W.	951	Langdon, F.J.	205, 206
Klimasara, A.	69	Krishna, M.B.	1013	Lange, R.	811
Klinger, D.L.	1838	Krishnappa, G.	1510	Langenbucher, V.	176, 177
Klosner, J.M.	535	Kroebel, W.	22	Lapini, G.	426
Klosterman, A.L.	666, 1948	Kroll, R.K.	1174	Laratta, A.	433
Klump, R.	1618	Kronauer, R.E.	1237	Large, J.B.	318

Larrabee, R.D.	1249	Levin, P.	735	Lottati, I.	146
Larsson, L.	407	Levy, A.	1159	Lotz, R.	1516
Lasagna, P.L.	156, 1999	Levy, D.A.	1079	Lotze, A.	974, 1354
Lau, J.C.	1168	Levy, S.	826	Louden, M.	1011
Laudien, E.	176	Lew, H.S.	229, 2041	Lowen, G.G.	1894
Laudiero, F.	1050, 1954	Lewis, A.B.	1230	Lowenadler, R.	1782
Laura, P.A.A.	539, 1339, 1444, 1651	Lewis, R.B.	167, 564	Lu, H.Y.	671
Laurenson, R.M.	418	Liang, C.Y.	909	Lu, Y.P.	340
Law, E.H.	1210	Libai, A.	537	Lubin, B.T.	2135
Lawrence, I.	1624	Liebe, R.	592	Lucas, J.G.	1862
Lazzeri, L.	382, 383	Lieberman, P.	511	Luco, J.E.	491, 802, 1078, 2151
Leadbetter, S.A.	639	Liebig, S.	1907	Lufrano, L.A.	703
Leasure, W.A., Jr.	840, 1052, 1218	Likins, P.	437	Luisoni, L.E.	1444
Leatherwood, J.D.	1190	Likins, P.W.	617	Lull, B.	1160
Leblois, L.C.	381	Lin, C.	1955	Lumsdaine, E.	544, 1961
Ledbetter, R.H.	211, 355	Lin, C.W.	1215	Lund, J.W.	412
Lee, E.H.	293	Lin, G.	1343	Lutes, L.D.	1751
Lee, H.S.H.	1211	Lin, G.F.	1909	Lutton, R.J.	360
Lee, I.K.	2098	Lin, H.-C.	532, 959, 1313	Luyties, W.H., III	798
Lee, J.	910	Lin, J.	247	Lynch, J.P.	1254
Lee, K.L.	1269	Lin, K.-H.	2027	Lynch, J.W.	167, 564
Lee, R.	1494	Lin, W.-H.	1847	Lyon, R.H.	877
Lee, S.H.	848	Lin, Y.K.	1296, 1297, 1298	Lysmer, J.	1059, 2185
Lee, S.M.	838	Lindberg, H.E.	38	Lytton, R.L.	1060
Lee, S.S.	1187	Lindgren, B.J.	371		
Lee, T.H.	706	Lindsey, L.G.	434	- Mc -	
Lee, T.T.	961	Ling, S.	1316		
Lee, W.N.	673, 674, 1429	Linscott, B.S.	1007		
Leehey, P.	874, 956, 1637	Linton, D.L.	638	McArdle, J.G.	1842
Lees, A.W.	445	Lister, T.A.	860	McCabe, M.W.	1741
Leffert, R.L.	601	Little, D.R.	191	McCallion, H.	1453, 1455
Leggat, L.J.	1005	Little, R.M.	223	McCann, J.C.	1998
LeGuilly, G.	629	Liu, C.H.	19	McCarthy, M.F.	265
Lehmann, E.J.	259	Liu, C.Y.	1317	McCleary, L.E.	459
Leipholtz, H.H.E.	486, 2142	Liu, J.T.C.	1816	McConnell, R.D.	882
Leist, T.	2106	Liu, R.	657	McDaniel, D.	578
Leleux, F.	455	Liu, S.-C.	466, 1680	McDaniel, S.T.	1916
LeMaitre, J.F.	615	Liu, Y.K.	128, 363	McDanien, T.J.	1243, 1266
Lennox, W.C.	244	Lo, D.L.C.	762	McDonough, J.R.	1562
Lenz, R.W.	556, 1778	Lo, H.	1033	McElroy, W.J.	1279
Leonard, B.R.	1002	Lobitz, D.W.	1144	McEvilly, T.V.	1267
Lepik, Ü.	1831	Lockwood, J.C.	682	McGarvey, J.H.	808
Leppert, E.L.	848	Lohmann, D.	99	McGehee, B.L.	60, 497
Lerner, E.	149	Loiseau, H.	805	McGeorge, R.	830
Leskovar, P.	1200	Lokken, E.C.	545	McGuire, D.P.	480
Lester, H.C.	1321	Lomas, N.S.	1650	McGuire, R.K.	2009
Letty, R.M.	2012, 2013	Longcope, D.B.	753	McHenry, R.R.	1254
Leung, C.M.	694	Longhouse, R.E.	819, 2022	McIvor, I.K.	1255, 2029
Levek, R.	1736, 1737	Longinow, A.	357	McKechnie, J.C.	1947
Leventhall, H.G.	760	Lord, H.W.	1662	McKeever, B.	1778
		Lorusso, J.J.	512	McKinlay, W.P.	1933

McLeod, R.W.	1641	Malyshov, V.S.	1090	Maurer, O.	488
McNamara, R.J.	2088	Mangiante, G.A.	872, 1990	Mayer, A.	1749
McNiven, H.D.	2113	Mann, R.L.	371	Mayer, W.G.	7
McQueen, A.A.	1713	Manning, J.C.	2194	Mayes, I.W.	445
McQueen, D.H.	767	Marangoni, R.D.	2139	Mayes, W.H.	167, 564, 1497
McQueen, D.H.	814	Marchertas, A.H.	588	Maymon, G.	536, 537
McWhannell, D.C.	248, 2128	Marciniak, T.J.	588	Maytum, B.D.	452
- M -					
Ma, D.C.-C.	1822	Marcuson, W.F., III	523	Mazumdar, J.	1154
Ma, S.M.	379, 1102	Margolis, D.	604	Mead, D.J.	82
Mabey, D.G.	1758	Markenscoff, X.	953, 1152	Mechel, F.P.	97, 140, 690, 691
Mabie, H.H.	741	Markho, P.H.	925	Medaglia, J.M.	635
MacAdam, C.C.	1220	Markuš, Š.	770	Medearis, K.G.	800
MacBain, J.C.	958	Marlotte, G.L.	388	Meerkov, S.M.	1535
MacDonald, J.	582	Maroney, G.E.	1278, 1469	Mehner, R.	2152
Machin, K.E.	1608	Marples, V.	1867	Mei, C.	1552
Macinante, J.A.	1690	Marsh, A.H.	2159	Mei, C.C.	850
Macintyre, S.A.	513	Marsh, J.C., IV	1368	Meier, G.E.A.	1319
Mack, R.J.	1495	Marshall, R.D.	1848	Meirovitch, L.	178, 254, 1038,
MacKay, A.	1427	Marshall, T.A.	1752	1065, 1656	
MacKay, J.F.W.	1911	Martin, C.R.	358	Meldrum, B.H.	1722
Macvean, D.B.	1707	Martin, D.J.	947	Melosh, R.J.	1757
Maddox, H.A.	2019	Martin, N.C.	1637	Melvin, P.J.	1884
Maddox, K.C.	2100	Martin, R.	1734	Memula, L.	1140
Madsen, N.F.	1422, 1423, 1896, 1898	Martins, R.A.F.	1330	Mengason, J.	1532
Maekawa, S.	1296, 1297	Martz, J.W.	2106	Mengi, Y.	2113
Maekawa, Z.	1486, 2077	Mason, V.	1573, 1661	Mente, L.J.	673, 674, 1429
Maestrello, L.	19	Masri, S.F.	2111	Mercier, O.L.	2049
Maezawa, S.	276	Massing, D.E.	1527, 1528	Metcalfe, R.W.	460
Magliozzi, B.	352, 353, 354, 1995	Masubuchi, M.	1473	Metzger, R.	625
Magrab, E.B.	955	Masuko, K.	1397	Meyer, A.	987
Mahabaliraja	1644	Masur, E.F.	762	Meyer, H.	718
Mahajan, K.K.	1683	Mather, C.E.	1542	Meyer, R.J.	2183
Mahalingam, S.	2, 648	Mathews, D.C.	1843	Meyer zur Capellen, W.	1631
Mahrt, K.H.	22	Mathews, D.E.	840	Miao, W.-L.	1784
Maidanik, G.	938, 1539	Mathews, F.H.	506	Michalopoulos, A.P.	593
Maiti, M.	1980	Matsuda, T.	1523	Michalopoulos, C.D.	2196
Majka, J.W.	639	Matsumoto, G.Y.	1116	Michimura, S.	1456
Majumdar, B.C.	1110	Matsumoto, M.	981	Miessen, W.	1617
Makay, E.	1008	Matsumura, M.	594	Mikulas, M.M., Jr.	1625
Makdisi, F.I.	2084	Matsuura, K.	1285	Milulcik, E.C.	454
Mall, S.	122	Matsuura, K.	2193	Milenkovic, V.	1633
Mallett, R.L.	293	Matsuzaki, A.	1775	Miles, A.W.	290
Mallick, D.V.	1983	Matsuzaki, Y.	1053	Miller, C.A.	703
Mallik, A.K.	82	Matta, R.K.	1385	Miller, R.D.	1174
Maloney, J.G.	508	Matthew, G.K.	1654, 1655	Miller, R.K.	1727
Malthan, J.A.	1597	Matthews, A.T.	966	Miller, T.D.	2076
		Matthiesen, R.B.	563	Mills, J.F.	157, 793
		Matthys, C.G.	1856	Milne, R.D.	2179
		Mattox, R.M.	1851	Milne, W.R.	575
		Matzen, V.C.	2083	Milordi, F.W.	1797
		Maunder, L.	284	Minagawa, S.	271

Mineck, R.E. 2164
 Minich, M.D. 1329
 Minner, G.L. 101, 1035
 Minto, R.F. 971
 Minton, P. 2131
 Mioduchowski, A. 1611
 Miramand, N. 455
 Mirza, J.F. 1312
 Mizra, S. 333
 Mishler, R.B. 145, 2163, 2197
 Mishoe, J.W. 2176
 Mishra, A.K. 5
 Misra, A.K. 87
 Mitchell, G.C. 337
 Mitchell, J.S. 1280, 1583, 1769,
 1928
 Mitchell, W.S. 1056
 Mitropolskij, J.A. 2044
 Mittendorf, S.C. 341
 Mixson, J.S. 1491, 1497
 Miyakawa, S. 1014
 Mizoguchi, K. 1823
 Mizuno, N. 41
 Mlakar, P.F. 859
 Modi, V.J. 87, 1647
 Moeller, T.L. 339
 Moes, H. 1451
 Mohraz, B. 35
 Moiseev, N. 2075
 Mojtabehi, S. 1185
 Molina, M.A. 303
 Mommessin 1305
 Mondkar, D.P. 1244
 Monfort, A. 1720
 Monroe, N.J. 646
 Montagnani, M. 196
 Montoya, L.C. 398
 Mook, D.T. 1144, 1885
 Moore, E.F. 943
 Moran, D.D. 1426, 1525, 1706
 Moran, M.J. 927
 Moravec, E.P. 1698
 Moreland, J.B. 971
 Morfey, C.L. 109, 1168
 Morino, L. 1253
 Morman, K.N., Jr. 1394, 1927
 Morris, N.F. 864
 Morris, P.J. 1168
 Morris, R.D. 729
 Morrison, D. 424
 Morrone, A. 829

Morrow, C.T. 469
 Mortell, M.P. 680
 Mortimer, R.W. 95, 1809
 Morysse, M. 768
 Moszee, R.H. 1807
 Mote, C.D., Jr. 647, 1147
 Motosh, N. 105
 Motsinger, R.E. 101
 Mott, K.J. 1665
 Moustafa, M.A. 86
 Mouzakis, T. 1138
 Mróz, Z. 1831
 Muehlbauer, G. 153
 Mueller, A.W. 24
 Mueller, M.W. 566
 Muir, T.G. 682
 Mukherjee, A. 1107
 Mukerjee, P.R. 1103
 Mukhopadhyay, A.K. 114
 Mulcahy, T.M. 1866, 2145
 Mulholland, G.P. 931
 Müller, H.W. 1663
 Müller, J. 317
 Muller, R.A. 586
 Munjal, M.L. 2061
 Muñoz, A., Jr. 1851
 Munson, B.R. 520
 Murakami, H. 2151
 Murakami, M. 1181
 Murata, S. 311
 Murotsu, Y. 997, 1192
 Murphy, J.R. 1072
 Murray, G., Jr. 1085
 Murthy, P.A.K. 126
 Murthy, V.R. 312, 990
 Murty, A.V.K. 1049
 Müsseler, P.M. 1367
 Mustain, R.W. 724
 Muszynska, A. 607
 Muto, S. 1401
 Mutyala, B.R.C. 200
 Myers, M.K. 1121, 1622
 Myles, M.M. 1836
 Myrick, S.T., Jr. 845

- N -

Nagamatsu, A. 1456
 Nagaraj, V.T. 1964
 Nagasaka, I. 1041
 Nagaya, K. 121, 1070, 1325, 1472,
 1649, 1801, 2112
 Nagel, R.T. 1843
 Naguib, M. 1865
 Nagy, K. 1436
 Nahavandi, A.N. 829
 Nair, S. 1104
 Naka, A. 1396
 Nakajima, S. 1814
 Nakamura, Y. 1929
 Nakano, M. 1164
 Nakayama, M. 1714
 Nakra, B.C. 78, 912
 Nalecz, A. 599, 1402
 Nam, C.H. 989
 Namba, M. 1308
 Narayana Raju, P. 1964
 Narkis, Y. 11
 Nash, A. 1196
 Nash, W.A. 326, 881, 962, 1332,
 1979
 Nataraja, R. 954
 Nathoo, N.S. 1869
 Natke, H.G. 723, 2053
 Nau, R.W. 534
 Naveh, B.M. 1289
 Nayfeh, A.H. 242, 952, 1118,
 1144, 1885, 2119
 Neily, D.W. 579, 1877
 Neise, W. 1001, 2023
 Nelson, D.V. 921
 Nelson, F.C. 36, 1897
 Nelson, H.D. 422, 1229
 Nelson, I. 1569
 Nelson, M.F. 1922, 2027
 Nelson, R.B. 695
 Nelson, R.C. 350
 Nelson, R.W. 1500
 Nemat-Nasser, S. 271
 Neshe, P.P. 1924
 Ness, D.J. 642
 Ness, H.B. 1787
 Nessler, G.L. 2100
 Neubauer, W.G. 21
 Neubert, V.H. 1953
 Neuhäuser, H. 1703
 Newman, M. 670, 1561
 Newman, R.A. 1028
 Ng, G.S. 1556
 Ng, S.-L. 443, 1036
 Nguyen, P.K. 1718

Nguyen, Y.T.	253, 669	Oden, J.T.	884	- P -	
Ni, C.C.	919	Oertel, H.	461		
Ni, R.H.	307	Ogino, S.	1812	Pace, C.E.	700
Niblett, T.	1672	Oh, K.P.	923	Packman, P.F.	1081
Nicholas, J.	805	Ohkami, Y.	437	Paidoussis, M.P.	295, 1135
Nicholas, J.C.	1449	Ohmata, K.	37, 342	Pallott, D.S.	862
Nichols, J.F.	1666	Ohta, M.	1759	Palmer, W.E.	1626
Nieberding, W.C.	1946	Ohtsubo, H.	448	Palmissano, R.R.	579
Niedbal, N.	1287	Ohyoshi, T.	2092	Pamidi, M.R.	1560
Niederer, P.	1393	Ojalvo, M.S.	665	Pamidi, P.R.	1560
Nigam, S.P.	2141	Okabayashi, K.	594	Pan, K.C.	2127
Nigm, M.M.	1694, 1695	Okada, T.	2008	Pandit, S.M.	2182
Nigul, U.	1761	Okada, Y.	1986	Pao, S.P.	2194
Nijim, H.H.	1298	Okah-Avae, B.E.	405	Papadakis, C.N.	110
Nilsson, A.C.	2037	Okawa, D.M.	130	Pappalardo, M.	1742
Nishida, S.	33	Okazaki, K.	117	Parekh, C.J.	1888
Nissim, E.	146	Okumura, A.	1043, 1240, 1241	Park, C.A.	226
Nobile, M.A.	1479, 1480	Oldham, G.A.	470	Park, K.C.	214
Noble, S.L.	1213	Oldham, K.	1465	Parkinson, A.G.	419
Nocilla, S.	1040	O'Leary, T.R.	265	Parry, D.L.	898
Nogami, T.	1925, 1926	Olhoff, N.	137	Parry, H.J.	1544
Nolan, D.	1776	Olsen, N.L.	1794	Parry, J.K.	1544
Noll, R.B.	1253	Olsen, W.A.	462	Patel, B.M.	183, 1031
Noll, T.E.	1785	Olson, D.A.	390	Patel, J.S.	1559
Nollau, R.	1889	Olson, M.D.	1151	Paterson, R.W.	59
Nonaka, T.	1295	Olson, R.M.	1023	Paton, J.A.	65
Noor, A.K.	949	Olsson, S.	1715	Patrickson, C.P.	1184
Nordlin, E.F.	1223	Olver, N.D.	1697	Pattabiraman, J.	328, 1645
Nordman, R.	430	Omori, Y.	1523	Patterson, W.N.	441
Norgan, R.F.	1438, 1595	On, F.J.	634	Paul, B.	1256
Norman, R.S.	1594	O'Neal, D.L.	1469	Pauls, L.	1024
Noronha, P.	732	O'Neill, J.	511	Pavic, G.	733, 1593
Norton, M.P.	1640	O'Neill, M.W.	80	Paz, M.	517, 518
Norwood, F.R.	1920	Oran, C.	763	Pearson, D.S.	1937
Novak, M.	1324, 1362, 1925, 1926	Orlandea, N.	1257, 2068, 2069	Pearson, J.	552
Novick, A.S.	1113	Orndorff, R.L., Jr.	936	Peckham, R.G.	1970
Nowinski, J.L.	301	Osinski, J.	2038	Peeken, H.	1636
Nuske, D.J.	373, 374	Orszag, S.A.	460	Pegg, R.J.	1995
Nuttall, S.M.	411	Oster, K.B.	1177, 1359	Penko, P.F.	1842
Nystrom, P.	2194	Ostergaard, P.B.	577	Pennick, H.G.	1827
- O -					
Obi, C.	237, 238	Ostrom, D.K.	1854	Penzes, L.E.	821
O'Brien, J.	107	Ott, H.	1619	Penzien, J.	299, 983, 1181
O'Callaghan, M.J.A.	326, 1332	Ottens, H.H.	1169	Peracchio, A.A.	975
Ochiai, Y.	386	Ottl, D.	1097	Perangelo, H.J.	1797
O'Connell, R.F.	855	Overgard, D.L.	837	Perlee, H.E.	1460
Odell, A.H.	2160	Owen, D.R.J.	595, 1330	Perlman, A.B.	867, 868
Odello, R.J.	1234	Owarz, A.	403	Perso, J.C.	201
		Özdemir, H.	1068	Perulli, M.	56
				Pestorius, F.M.	345, 681
				Peterka, J.A.	800
				Peters, A.J.	1106

Petersen, E.	1629	Pope, L.D.	108	Rader, P.	1233
Peterson, A.J.	1875	Pope, R.L.	2073	Radhakrishnan, V.M.	1112
Peterson, E.L.	1948	Popeck, R.A.	546	Radhamohan, S.K.	1980
Petrauskas, C.	1315	Popov, P.	1102	Radochia, J.P.	2036
Petre, A.	555	Popovici, A.	801	Radovich, N.A.	855
Pett, R.A.	1927	Popp, K.	1380, 1603	Radovich, V.G.	392, 393
Petyt, M.	1988	Popp, L.E.	688	Raffy, P.	815
Pfützner, H.	72	Popplewell, N.	1911	Ragab, S.	1961
Phadke, M.S.	188	Porter, F.L.	830	Raghavan, R.	750
Phelps, H.N., Jr.	1470	Portillo Gallo, M.	1360	Raila, D.S.	165
Philbert, M.	721	Posey, J.W.	527, 1321	Rajagopal, P.	148
Philippin, G.	486	Possa, G.	426	Rajan, G.	1130
Phillips, J.W.	1273, 1304	Potter, D.K.	974	Raju, I.S.	914
Phoa, Y.T.	150	Potter, R.C.	2157	Raju, K.K.	118, 914, 1155
Piazzoli, G.	562, 2004	Powell, A.	1996	Rakowski, W.J.	1598
Pickles, J.M.	1599	Powell, G.H.	1244	Ramachandran, J.	126
Pierce, A.D.	917, 970	Powell, H.N.	61	Ramachandran, J.	1336
Pierce, G.A.	312	Prabhakara, M.K.	1820	Ramamurti, V.	328, 1645, 2059
Piersol, A.G.	621	Prabhu, B.S.	1109, 1452	Ramanathan, R.	1112
Pierson, W.D.	1755	Prasthofer, P.H.	474	Ramberg, S.E.	930
Pierucci, M.	1420, 1421, 1895, 1899	Prause, R.H.	1212	Ramey, D.G.	2019
Pigott, R.	1148	Prendergast, J.D.	1679	Ramkumar, R.L.	1270
Pike, A.L.	1051	Price, I.R.	932	Rammerstorfer, F.G.	541
Pilkey, W.D.	180, 223, 576, 1022, 1575	Price, P.	1234	Ramu, S.A.	740
Piltz, E.	1724	Price, W.G.	225, 1291	Rand, D.A.	1418
Pincus, G.	1685	Priscu, R.	801	Randall, R.B.	1585
Pinnington, R.J.	1988	Privitzer, E.	357	Rangacharyulu, M.A.V.	1883
Piotrowski, E.	151	Prodanovic, A.	2094	Rangaiah, V.P.	1953
Pister, K.S.	1268	Prodoehl, R.F.	1223	Ranlet, D.	1828
Pitts, L.E.	7	Protonotarios, J.N.	2171	Rao, B.V.A.	1452
Piziali, R.L.	1405	Prydz, R.A.	757	Rao, G.V.	118, 914
Plecnik, J.M.	1614	Pugh, C.A.	2010	Rao, H.V.S.G.	75
Pleec, G.	2018	Purcell, W.E.	1987, 2155	Rao, J.S.	1054
Plona, T.J.	7	Puri, A.	2135	Rao, N.S.	1450
Ploner, B.	988	Pusey, H.C.	2104	Rao, N.S.V.K.	1272
Plumbee, H.E., Jr.	1168	Putman, C.B.	1370	Rao, P.N.	740
Plunkett, R.	904, 1074, 1075, 2107	Putman, W.F.	973	Rao, S.S.	1475
Pocha, J.J.	1231	Putnam, T.W.	156	Rasmussen, M.L.	1432
Poelaert, D.H.L.	610	- Q -			
Poizat, M.	481	Queller, J.E.	164	Rathe, E.J.	1513
Polak, E.	1268	Quinlan, P.M.	326, 1332	Ratz, A.G.	505
Polak, E.J.	1568	Quinn, B.E.	397	Rauch, M.	1439
Poli, C.	991	- R -			
Pollack, I.	2078	Radaj, D.	2031	Rawlins, A.D.	29, 1352
Pollack, J.L.	1946	Radcliffe, K.S.	170	Rawls, E.A.	640
Pollmann, E.	425				
Pombo, J.L.	1444	Rebora, B.		Ray, A.	583
Poole, J.H.B.	760	Reboulet, C.		Ray, D.	1268
Poon, D.T.	1647	Reddy, C.T.		Read, P.L.	1526
		Reddy, C.T.		Rebora, B.	124
		Reddy, C.T.		Reboulet, C.	615
		Reddy, C.T.		Reddy, C.T.	884
		Reddy, N.N.		Reddy, N.N.	547
		Reed, C.		Reed, C.	323

Reed, W.E.	2055, 2056	Röhrle, M.	387, 1015	- S -
Reed, W.H., III	1798	Rom, J.	52	
Reeves, C.W.	404	Romeo, D.J.	1025	
Régnault, G.	1284	Romilly, N.	240	
Reifsnider, K.L.	274	Roos, R.	1498	
Reismann, H.	116	Root, R.M.	996	
Reissner, E.	1951	Roriston, G.	1932	
Reiter, W.F., Jr.	395, 1115, 1398	Rosati, V.J.	907	
Rekos, N.F., Jr.	1843	Rosen, A.	967	
Reshotko, M.	1842	Rosenblueth, E.	2099	
Rettig, H.	528	Roshala, J.L.	688	
Reyna-Allende, M.	1408	Ross, C.A.	476	
Reynolds, D.D.	1355, 1356, 1357	Rossall, A.W.	1913	
Reynolds, G.G.	1205	Rosettos, J.N.	1017	
Rice, C.G.	1743, 1744	Rossini, T.	426	
Rice, E.J.	1035, 1123	Rostafinski, W.	30	
Rice, G.	769	Roth, C.A.	440	
Richards, E.J.	661	Roth, H.	1437	
Richardson, D.A.	1351	Rousselet, J.	1327	
Richardson, G.N.	135, 1269	Rovetta, A.	1692	
Richardson, M.	727	Roy, K.P.	719	
Richardson, R.S.H.	1577	Royle, P.	1489	
Richter, R.	1908	Royster, L.H.	28	
Rider, R.W.	1684	Rubayi, N.A.	79	
Riegel, C.	301	Rubin, M.N.	1667	
Rieger, N.F.	415, 1419	Rudisill, C.S.	1765	
Rigbi, Z.	1762	Rudny, D.F.	346	
Riffel, R.E.	1113	Rueckemann, O.	67	
Riganti, R.	1416	Rueter, F.	717	
Rizzi, P.	745	Ruff, J.H.	1696	
Robbins, D.H.	1369	Ruhlin, C.L.	554	
Roberson, J.A.	891	Rulf, B.	1260	
Roberts, J.B.	652, 1067, 1288	Rup, W.	1024	
Robertson, D.H.D.	1933	Russell, A.J.	1592	
Robinson, R.R.	357	Russell, G.A.	2156	
Robinson, S.M.	15	Russell, J.J.	88, 1447, 1803	
Robson, J.D.	1707	Russell, R.H.	668	
Rocke, R.D.	17	Rusu, O.	1976	
Rockwell, T.H.	734	Rutenberg, A.	1157	
Rodal, J.J.A.	1522	Ruter, G.	391	
Rodeman, R.	753	Rutgersson, O.	1715	
Rodrigo, P.	615	Ruud, F.O.	1207	
Roe, G.E.	1403	Ryden, C.V.	500	
Roessel, J.M.	492, 1852	Ryder, M.O.	1018, 1019, 1020,	
Rogers, C.B.	741		1388, 1389	
Rogers, C.R.	1904	Ryland, G., II	1656	
Rogers, J.L., Jr.	1552, 1561	Rylander, H.G.	845	
Rogers, K.J.	1700	Rylander, R.	212, 1512, 1746	
Rogers, P.	.964, 965	Ryneveld, A.D.	2167	
Rogers, R.J.	1133			
Rohde, S.M.	923			

Schmidt, L.C.	1818	Sexton, J.S.	2179	Shum, K.	193
Schmidt, W.	1439	Seybert, A.F.	204, 1675	Shutt, M.D.	1431
Schmitz, F.H.	27, 361	Seymour, B.R.	680	Sidar, M.	349, 858
Schneider, A.J.	292	Shaaban, S.H.	962, 1753	Sidell, R.S.	1328
Schoen, B.	2165	Shafer, B.P.	296	Sidwell, K.	1173
Scholes, W.E.	464	Shaffer, G.M.	402	Sierakowski, R.L.	476
Scholl, H.F.	167, 564	Shah, H.H.	1850	Silver, M.L.	1521
Scholl, R.E.	168	Shah, M.P.	392, 393	Silver, W., II	580
Schomer, P.D.	32, 2014	Shah, P.C.	687	Simandiri, S.	1616
Schrantz, P.R.	1876	Shaker, B.S.	2119	Simiu, E.	1848
Schroder, E.	285	Shamie, J.	2173	Simkins, T.E.	2086
Schubert, K.-H.	2065	Shampine, L.F.	753	Simmons, H.	1690
Schultz, T.J.	64	Shanks, R.E.	24	Simmons, H.R.	608, 893
Schulz, G.	1839	Shantaram, D.	595	Simmons, J.M.	1613
Schulz, R.	1723	Shanthakumar, P.	1964	Simmons, P.E.	444
Schuring, D.J.	605	Shapton, W.R.	1007	Simpson, A.	1314
Schuring, D.S.	1476	Sharp, B.H.	217, 842, 843	Simpson, B.A.	979
Schuster, G.M.	287	Sharp, J.C.K.	129	Simpson, M.A.	1991
Schutt, D.W.	69	Sharp, R.S.	841	Simpson, W.J.	1116
Schweiger, W.	1227	Shaw, L.M.	1442, 1942	Singer, J.	967
Schweitzer, G.	811, 1379	Shaw, R.P.	710, 1335	Singh, J.P.	1853
Schwer, L.	1126	Shearer, J.C.	1565	Singh, K.P.	696, 1967
Schwerdtfeger, H.	425	Shearer, J.E.	1026	Singh, M.C.	5
Sciara, J.J.	1556	Sheer, R.E., Jr.	1863	Singh, M.P.	1311
Scofield, K.E.	513	Sherif, M.A.	493	Sinha, P.K.	1146
Scott, J.E.	1392	Shevell, R.S.	1496	Sinha, S.C.	851
Scott, J.N.	1002	Shibata, H.	822, 1137	Sinha, S.K.	1344
Scott, N.	689	Shields, F.D.	1943	Siorek, R.W.	348
Scott, R.A.	77	Shigeta, T.	1137	Siskind, D.E.	170, 1781
Scott, W.E.	773	Shih, S.	1767	Sjöstedt, E.	1746
Scott, W.R.	1923	Shimizu, H.	1042	Skaistis, S.J.	344
Seed, H.B.	1059, 1766, 2185	Shimizu, T.	1508	Skale, S.R.	842
Seehra, M.S.	288	Shimovetz, R.M.	389	Skingle, C.W.	1670
Seeliger, A.	1541	Shin, Y.S.	1375	Skjeltorp, A.T.	266, 267, 268,
Segal, D.J.	399	Shinozuka, M.	1676	269, 270	
Segenreich, S.A.	745	Shirai, K.	1748	Skoglund, G.R.	523
Seginer, A.	52	Shiraishi, N.	981	Skop, R.A.	930
Seiffert, U.	473	Shirakawa, K.	1823	Skudrzyk, E.J.	2074
Seiler, J.P.	504	Shiraki, K.	822	Sloan, I.H.	449
Seiner, J.M.	2194	Shirota, K.	438	Small, W.M.	2136
Sekiguchi, T.	1299, 1323	Shivashankara, B.N.	2003	Smallwood, D.O.	728
Sen, S.K.	1139	Shoemaker, C.O.	840	Smith, C.B.	585, 827
Seneor, R.	13	Shoemaker, N.E.	1019, 1020,	Smith, C.C.	861
Senoo, Y.	1203		1388, 1389	Smith, C.S.	337
Sensburg, O.	1354	Shore, S.	1845, 2110	Smith, D.A.	1832
Seo, K.	1390	Short, S.A.	706	Smith, D.L.	389
Setiawan, B.	333	Shovlin, M.D.	1993	Smith, E.H.	924
Seto, K.	1000	Shrader, J.T.	216, 1217	Smith, G.R.	1391
Severn, R.T.	645	Shreve, J.C.	1441	Smith, J.B.	364
Sevin, E.	510	Shryock, R.A.	2034	Smith, J.C.	1312
Sewall, J.L.	639	Shukla, D.K.	593	Smith, J.D.	304

Smith, J.H.	2042	Steers, L.L.	398	Suryoutomo, H.	1139, 1829
Smith, L.M.	1787	Stein, R.J.	1662	Susemihl, E.A.	1339, 1444
Smith, M.J.T.	831	Steinberg, D.S.	1337	Sutterlin, M.W.	917
Smith, R.S.	1773	Stelma, J.L.	2166	Sutton, M.A.	885
Smith, S.	502, 725	Stematiu, D.	801	Suzuki, S.	1861
Smith, T.E.	39	Stephens, D.G.	167, 564	Suzuki, S.I.	1648, 1653, 1711
Smith, T.J.B.	1129, 1361	Stephens, J.E.	1301	Suzuki, T.	386
Smith, W.F.	786	Stephens, W.B.	639	Svalbonas, V.	123
Snediker, D.K.	1282	Stephenson, J.E.	28	Svetlitsky, V.A.	2132
Snowdon, J.C.	1290, 1350, 1371, 1479, 1480	Stepniewski, W.Z.	794	Swamy, S.T.N.	1452
Snyder, J.E., III	1300	Stevenson, J.D.	198	Swaney, T.G.	1755
Sobczyk, K.	1707	Stevenson, J.R.	1844	Sweet, A.L.	859
Soedel, W.	190, 332	Stewart, J.	319	Symmons, G.R.	1047
Sofronie, R.	849	Stewart, J.S.	789	Syring, R.P.	1755
Sogabe, K.	1137	Stewart, N.D.	184	Szechenyi, E.	698
Solecki, R.	111	Stewart, R.M.	369	Szu, C.	458
Somers, A.E.	1551, 1740	Still, P.W.	1083	- T -	
Soni, S.R.	1981	Stimler, F.J.	1483	Tag, I.	544
Soong, T.T.	358	Stinchcomb, W.W.	274	Tagata, G.	1612
Soovere, J.	1729	Stockdale, W.K.	442	Tai, J.	891
Sopher, R.	606	Stoker, J.R.	1223	Takamatsu, Y.	1204
Sörensen, S.	212, 1512	Stone, B.J.	2179	Takasu, ^	280
Soroka, W.W.	241	Stott, J.D.	1780	Takatsu, i.	315
Sorsdal, S.	289	Stouder, D.J.	1998	Takei, A.	594
Sotomura, K.	823	Stouffer, D.C.	2100	Takemiya, H.	1751
Sozen, M.A.	1347	Stoykovich, M.	828	Takemori, T.	823
Spahr, H.R.	163	Strauss, A.M.	978	Takeyama, H.	1299, 1323
Spanos, P.T.	1566	Strenkowski, J.	1575	Takizawa, H.	644
Sparks, C.R.	1968	Stricklin, J.A.	852	Taleb-Agha, G.	1919
Spencer, R.	794	Stringas, E.J.	2163, 2197	Tam, D.S.F.	1066
Sperry, W.C.	1247	Strona, P.O.	383	Tanaka, H.	1960
Spillman, J.J.	1582	Stroud, R.C.	502, 725	Tanida, T.	47
Spitzig, W.M.	521	Strout, F.G.	1994	Taniguchi, O.	865, 866
Spokas, J.J.	1235	Struessel, D.	1736, 1737	Tanna, H.K.	1071, 1167
Srichatrapimuk, T.	2149	Stuart, A.D.	2074	Tanner, R.B.	1021
Srinivas, S.	589	Stüber, C.	1010	Taoka, G.T.	657
Srinivasan, K.	2181	Stühlen, B.	1277, 1940	Tapia, G.A.	605
Srinivasan, P.	1883	Stusnick, E.	144	Taylor, A.S.	558
Stachiw, J.D.	1716	Styles, D.D.	453	Taylor, C.M.	924
Stachura, V.J.	170, 1781	Subramanian, A.K.	1709	Taylor, D.L.	1404
Stahle, C.V.	631, 634	Subramanian, R.	10, 1537	Taylor, R.F.	149
Stakolich, E.G.	1002	Subramanian, T.L.	810	Taylor, R.L.	249, 1882, 1957
Stammers, C.W.	1364	Sueoka, A.	1042	Taylor, S.M.	1745
Stamp, A.P.	685	Suggs, C.W.	364, 2176	Tein, Y.	746
Standlee, K.G.	1357	Sugimoto, N.	1886	Temkin, S.	694
Stargardter, H.	1458	Sullivan, J.W.	1675	Tendorf, Z.A.	116
Starnes, R.B.	2019	Sullivan, T.J.	191	ten Wolde, T.	736
Stea, W.	2066	Sun, C.T.	113, 538, 913, 950,	Teoh, L.S.	1605
Stearman, R.	1840	Sundararajan, C.	313, 1638		
Steenken, W.G.	1205, 1807	Surowiec, M.W.	1033		

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242

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1 - 78

DDC

Tepper, F.R.	1894	Tordion, G.V.	2123	Utz, W.R.	246
Teraoka, G.M.	642	Tosh, J.T.	1410		
Terauchi, Y.	103, 1127	Totos, K.	207		
Tesar, D.	1654, 1655, 2126	Townsend, J.L.	1499		
Tesch, W.A.	1807	Townsend, M.A.	1081		
Teschner, W.	2192	Traexler, J.F.	2060	Vaicaitis, R.	230, 1491, 1676
Tessarzik, J.M.	222, 224	Tran, C.T.	806	Vaidya, P.G.	1117
Testa, F.J.	1266	Traybar, J.	1293	Vaish, A.K.	830
Tester, B.J.	526, 759	Trifunac, M.D.	1078, 1275	Valathur, M.	1850
Thaller, R.E.	552	Triggs, T.J.	1687	Valdivieso, J.B.	2095
Thayer, W.J.	90, 922	Troeder, C.	1636	Valera, J.E.	2185
Thien, G.E.	203, 1216	Trogdon, S.A.	520	Vallet, M.	1518
Thoma, J.	2153	Trompette, P.	598	Valliappan, S.	2098
Thomas, C.B.	415	Troppens, D.	317, 1931	Van Atta, F.A.	372
Thomas, C.R.	76	True, H.C.	2000, 2012, 2013	Vance, J.M.	1057
Thomas, D.W.	597	Trumpler, P.R.	2101	Van Dao, N.	4, 1077
Thomas, E.	2065	Truppat, V.	951	VandeGriff, D.G.	508
Thomas, J.	743, 1292, 1309	Tsai, C.F.	1059, 2185	van der Burgh, A.H.P.	651
Thomas, T.R.	1128	Tschirner, G.	1802, 1966	van Dijk, J.S.C.	179
Thompson, A.G.	1490	Tseng, G.	2066	Van Dixhoorn, J.J.	1258
Thompson, A.G.	2032	Tso, W.K.	1157	Vance, J.M.	221
Thompson, B.S.	935	Tsui, C.Y.	1124	Vance, O.L.	141
Thompson, D.E.	2075	Tsui, Y.T.	1815	Vankirk, J.	181
Thompson, R.A.	1201	Tsujimoto, Y.	311	Van Kuren, R.C.	1392
Thompson, R.R.	1400	Tsushima, Y.	41	Van Laningham, F.L.	897
Thompson, W., Jr.	1910	Tucker, A.J.	938	van Leeuwen, H.	1451
Thomson, R.G.	1501	Tully, N.	926	Vanmarcke, E.H.	171, 172, 173, 174, 175
Thomson, W.T.	1246, 1872	Tung, C.C.	1312	Vann, W.P.	2042
Thornton, W.R.	189	Turula, P.	1866, 2145, 2150	Vannoy, D.	180
Tibbets, J.G.	547	Tustin, W.	1777	Van Nunen, J.W.G.	1236
Tichenor, D.R.	774	Twomey, W.	806	Van Schaick, T.E.	582
Tichy, J.	719			Vargas, L.M.	1878
Tindle, C.T.	44, 685			Varterasian, J.H.	1400
Ting, L.	19, 457, 1046			Vasudevan, R.	1112
Tiwari, R.N.	10, 1537			Vaucheret, X.	54
Tobe, T.	315, 2125			Vedrenne, M.	481
Tobias, S.A.	186, 1373, 1695			Veit, I.	70, 1944, 1945
Toda, H.	47, 272			Venema, T.	1521
Toda, S.	1822			Venetos, M.A.	512
Toda, Y.	438			Vepa, R.	1580
Todd, L.W.	1086, 1283			Vér, I.L.	1088, 1836
Tokarzewski, S.	477			Vernet, M.	1518
Tolani, S.K.	17			Vialatoux, P.	481
Tominari, N.	1000			Viets, H.	53
Tomizuka, M.	571			Vigner, F.R.	614
Tondl, A.	2045			Vigran, T.E.	289
Tong, P.	1012, 1017			Viksne, A.	986
Tonin, R.	1799			Villasor, A.P., Jr.	812
Tonndorf, J.	164			Vincent, R.J.	210
Tool, A.R.	1186			Virchis, V.J.	846
Topper, T.H.	486				

- U -

Visnapuu, A.	1488	Webb, W.W.	65	Wilby, J.F.	2000
Viswanathan, S.P.	1811, 1856	Webby, J.	976	Wilby, P.G.	2174
Vitelleschi, S.	1818	Weber, B.	1889	Wilcock, D.F.	1434, 1448
Vogel, W.H.	938	Weber, H.	718	Wilcox, D.J.	1913
Vogt, R.H.	21	Webster, R.L.	89	Wild, R.E.	1219
Voigtsberger, C.A.	787	Wegmann, R.	2114	Wilhelm, M.R.	1194
Volcy, G.C.	1714	Wegscheid, E.L.	786	Wilkinson, J.P.D.	826
von Cremer, L.	784	Wehrfritz, T.J.	1892	Wilkinson, K.	149
Vonnemann, G.	1858	Weinstock, H.	1211	Wilkinson, T.L.	316
Von Rosenberg, D.U.	128	Weiss, F.	2071	Willcox, M.G.	1349
Voorhees, C.R.	862	Weiss, G.H.	1912	Williams, C.J.H.	628
Vukelich, S.I.	101	Weiss, R.	235	Williams, D.R.G.	1433
		Weisshaar, T.A.	1270	Williams, J.	57, 58, 1900
		Weissman, S.	2066	Williams, J.L.	2089
		Weissner, R.	473	Williams, R.	1934
		Welaratna, S.R.	1587	Williams, R.S.	274
Wachel, J.C.	321, 946, 1968	Welford, G.D.	362	Williams, S.W.	681
Wada, B.K.	456, 848	Wellford, L.C., Jr.	334, 884	Williams, V.	781
Wada, H.	1265	Wells, W.R.	1120, 1320	Willis, C.M.	1497
Wade, S.R.	1485	Wellstead, P.E.	373, 374	Willis, T.	193
Wagner, H.	2059	Wenig, E.	1619	Willshire, W.L., Jr.	1989
Wales, D.R.	1453, 1455	Werner, V.A.	1221	Willsky, A.S.	2102
Walker, J.A.	751	Wesler, J.E.	1062	Wilson, D.M.	139
Walker, J.G.	1515	Wesley, D.A.	706	Wilson, G.J.	96, 929
Walker, J.S.	1273	West, L.R.	630	Wilson, H.E.	1503
Waller, H.	951	Westin, R.A.	2087	Wilson, J.C.	1870
Walsh, M.J.	1016	Westine, P.S.	475	Wilson, J.F.	520, 839, 1846
Walter, M.J.	1794	Weyer, R.D.	231	Wilson, L.O.	1302
Walter, W.W.	415	Whaley, P.W.	910	Wilson, W.R.D.	1507
Walther, R.	93, 94	Wheeler, P.	1399	Wiltzsch, M.	102
Wambsganss, M.W.	697, 1375, 1435	White, C.W.	452	Windett, G.P.	384, 385
Wang, B.P.	180	White, J.W..	1958	Winer, W.O.	303
Wang, C.	1171	White, K.C.	156	Winget, J.M.	754
Wang, H.T.	2064	White, M.F.	731, 1871	Winkler, C.B.	1477
Wang, J.C.F.	1863	White, R.G.	731, 2017	Winn, L.W.	713, 715, 1306, 1448
Wang, K.S.	1117	White, R.P., Jr.	2116	Winsor, F.J.	600
Wang, T.M.	1301	White, R.W.	166	Wisler, D.C.	1530
Warburton, G.B.	1981	White, W.	2098	Witmer, E.A.	1522
Ward, H.S.	2058	Whitehead, D.S.	928	Wittlin, G.	1172
Ward, P.	2121	Whitesides, J.L.	1242	Wohlrab, R.	1615
Ward, W.D.	26	Whitfield, E.L.	51	Wohltmann, M.	618
Warden, D.A.	948	Whitham, E.M.	570, 1345	Wolf, D.F.	163
Warwick, J.E.	435	Whitman, A.B.	1303	Wolf, J.A., Jr.	1922
Wassmann, W.W.	584	Whitman, R.V.	797, 1249, 1919, 2171	Wolf, J.P.	124, 707
Watanabe, T.	276	Whitney, J.M.	950	Wolf, S.N.	45
Waters, D.M.	880	Wickens, A.H.	2026	Wolfe, S.H.	494
Watson, E.E.	291, 908	Wiegand, V.G.	720	Wölfel, H.	785
Watson, M.L.	1246	Wierzbicki, T.	9, 916, 1952	Wong, C.	437
Wazniak, J.A.	1942	Wiggins, J.H.	1180	Wong, H.L.	491, 1078
Weaver, D.S.	1135, 2137	Wiland, J.H.	366	Woo, J.L.	302
				Woo, T.-H.	1730

Wood, A.D.	1821	Yamakawa, H.	1043, 1240, 1241	Young, J.W.	873
Wood, R.	1937	Yamamoto, T.	1041, 1264, 1712	Young, M.E.	1250, 1251
Woodcock, D.L.	560	Yamamoto, Y.	448, 1285	Young, R.K.	2136
Woodford, D.J.	416	Yamamura, H.	1800	Younger, F.C.	1872
Wooding, J.C.	1564	Yang, J.	783	Younghans, J.L.	191
Woodward, J.H.	141	Yang, J.C.S.	478	Yousri, S.N.	1824, 1825
Woodward, R.P.	220, 1862, 2021	Yang, J.Y.	331	Yuan, C.	1959
Woolf, A.	294	Yang, T.Y.	329, 1033		
Wormley, D.N.	1300, 1328	Yao, J.T.P.	169		
Worsfold, J.H.	445	Yashima, S.	1960	- Z -	
Wright, C.G.	1930	Yasuda, K.	1041, 1264		
Wright, E.W.	1179	Yates, P.E.	1527, 1528	Zabukovec, C.	756
Wright, J.W.	901	Yee, H.C.	1099, 1238	Zainea, B.	1098
Wright, S.E.	846	Yeh, T.T.	1982	Zak, A.R.	1819
Wu, C.L.S.	1632	Yen, D.H.Y.	1466	Zakkay, V.	457
Wu, D.-L.	1766	Yen, J.G.	1856	Zaman, F.D.	1821
Wu, J.J.	298, 2051	Yen, N.	1237	Zara, J.A.	621
Wu, S.M. 185, 188, 810, 2054, 2182		Yeow, K.W.	1911	Zarda, P.R.	1558
Wulkau, R.	2016	Yeowart, N.S.	1913	Zaschel, J.M.	2007
Wünsch, D.	1541	Yew, C.H.	40	Zienkiewicz, O.C.	595
Wuzyniak, J.A.	1442	Yin, S.K.	2190	Zimmer, A.	2031
Wyman, H.J.	1901	Yonemoto, J.	1750	Zimmermann, T.	124
Wyskida, R.M.	578, 1193, 1194	Yonetsu, S.	1508	Zinn, B.T.	227
		Yoneyama, T.	33	Zisling, D.	2008
		Yoshida, Y.	1401	Zobrist, G.J.	1902
		Yoshimura, M.	2180	Zockel, M.	1623, 1691
		Young, A.M.	286	Zorumski, W.E.	759
Yamada, M.	823	Young, C.J.	143, 969	Zorzi, E.S.	1229
Yamada, T.	1748	Young, F.J.	1468	Zuckerwar, A.J.	905
Yamaguchi, S.	1759	Young, J.	1938	Zuziak, R.J.	1876
				Zwaan, R.J.	1498
				Zwick, J.W.	101

- Y -

ANNUAL SUBJECT INDEX

- A -												
Absorbers (Materials)												
140	1481	1482										
			1836	767								
				1987								
Accelerometers												
	513											
		1056										
		1596										
Acoustic Absorbers												
			1987									
Acoustic Absorption												
690	131	872	1943									
760	691			1915	2156	97	788	1259				
1990	871			2155			988					
Acoustic Arrays												
	1742											
Acoustic Attenuation												
use	Acoustic Absorption											
Acoustic Diffraction												
	2074											
Acoustic Excitation												
1231	2062	1313	94	535	166	167	498	1739				
2161			1824	1825	536			1738				
Acoustic Fatigue												
2161												
Acoustic Filters												
451												
Acoustic Holography												
291			2055	2056			738					
							908					
Acoustic Impedance												
30	1122	33	34		526			759				
				1914								
Acoustic Insulation												
230		634		1836								
Acoustic Liners												
use	Acoustic Linings											
Abstract												
Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197
Volume 9												
Issue	1	2	3	4	5	6	7	8	9	10	11	12

Aerodynamic Loads											Aircraft Noise (Continued)						
1580 1401											1841 1352 1843						
											2001 1492 1993						
											2021 1842 2003						
Aerodynamic Response											1992						
use Aerodynamic Stability											2002						
Aerodynamic Stability											Aircraft Propellers						
501 1862	2174		1176		1408	1959					1261						
Aeroelasticity											Aircraft Seats						
											107						
Agricultural Machinery											Aircraft Vibration						
371 372			1696								1794 1175 1786 1787						
											2004 2005 1796						
Air Bags (Safety Restraint Systems)											Aircraft Wings						
1250 1251	213		365		2188						150 1671 1672	74	55	1236		1498 1669	
1391			1025								1670		1554	555			
Air Bags (Soft Landing)											1840		2164	1785		2165	
											1484						
Air Blast											Air Cushion Landing Systems						
170 1502			1754								1569						
Air Compressors											Airfoils						
use Compressors											1580						
Air Conditioning Equipment											Airframes						
100			1624								817	1492	1553	554		156	
											1844						
Aircraft											2164						
50 151	152	553	674	145	146	57	68	149			Airport Noise						
350 351	562	673	794	355	556	557	148	349			use Aircraft Noise						
560 551	1502	1173	974	1175	1496	1297	558	559			and Airports						
790 561	2042	1253	1174	1755	1546	2007	858	1169			Airports						
910 2161	2162	1353	1354	1795	2006						880 661						
						1668					1170						
Aircraft Engines											Ammunition						
	1843	1844	975	886	757						270		1574		266 267 268 269		
Aircraft Equipment											Amplitude Analysis						
552			1736	1737	1738	1739					1336						
Aircraft Landing Areas											Analog Simulation						
1171											1540						
Aircraft Noise											Analog Techniques						
60 161	62	153	24	25	26	27	58	159			1935						
160 191	162	353	154	155	56	157	158	549			Anechoic Chambers						
550 671	352	663	354	565	156	167	548	659			1591 1442						
660 791	462	793	564	905	1496	497	1168	1999			56						
880 1051	662	1493	1494	1495	1546	547	1998	2159			Anisotropic Properties						
1170 1441	672	1543	1674	1675	1996	717	2158				use Anisotropy						
2000 1461	722	1673	1744			1167											
2160 1491	792	1743	1994			1247											

Abstract

Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197

Volume 9

Issue	1	2	3	4	5	6	7	8	9	10	11	12
-------	---	---	---	---	---	---	---	---	---	----	----	----

Abstract

Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197

Volume 9

issue 1 2 3 4 5 6 7 8 9 10 11 12

Beams-Columns		Boats										
1740	763	1874	495									
Bearings												
1770	521 1082	714 715	716 1617	369								
	811 1112	924 1615	896 1957	1449								
	1541 1282		1206									
	1632		1806									
Bells		Bolts										
	822		1635									
Bernoulli-Euler Method												
910	2111 1602											
1100												
1300												
Bernoulli Theory												
		2109										
Bibliographies		Bones										
1250	1251 252	663 664	1735 1546	1547 1548	259							
	1673		1905 1826	2057								
			2055 2056									
Biomechanics		Booster Rockets										
		640 1232										
	978											
Blades		Boundary Condition Effects										
1960	721 522	93 94	756 1807	928 869								
	1113		755 846	2147 1308	929							
			1115 1456	1458 1459								
			1205 1946	1709 1769								
				1809 1959								
Blast Effects		Boundary Value Problems										
270		1880 1881	752									
2150		2051		235 1266								
Blast Loads		Box Beams										
511		1952										
	1429	Box Type Structures										
		1721										
Blast Resistant Construction		Brakes (Motion Arresters)										
use	Blast Resistant Structures		396									
Blast-Resistant Design		Braking Effects										
use	Blast Resistant Structures		1526									
Blast Resistant Structures		Branch Mode Techniques										
470 471 472	1755 2066	198	805 806									
1382												
Blast Response		Branched Systems										
170	1295		648									
Blowers		Bridges										
1510		1520	563 664									
		1790	795 1236									
		2110	1275 1846									
			1845									
Abstract		Bridging										
Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197			849									
Volume 9		Buckling										
Issue	1	2	3	4	5	6	7	8	9	10	11	12

Circular Plates	1340	1651	914	1155	1976	117	118	777	1147	Complex Structures	641	485											
	2011																						
Clutches										Component Mode Analysis	341	1904											
											136	1169											
Coal Handling Equipment										Component Mode Synthesis	641	822	853	1414	1428	429							
											1413				1159	1879							
Collapse										Composite Materials	1270	271	272	273	274	325	297	689					
use											1760		283	1074	1605		477						
Collision Research (Aircraft)														1923	1334								
use															1604								
Collision Research (Automotive)										Composite Structures	340				1075	766	538	939					
	1020	391	392	293	214	1245	1016	1017	208	209		950											
	1250	1021	542	393	394	1255	1126	1367	688	1019													
	1390	1251	1022	473	834		1366	1387	1018	1369	Composites												
							1391	1392	1023	1024		2186	1527	1368	1389								
									1922	1223	1254		1757	1388	2029								
													2027	1528	2189								
													2028										
													2188										
Collision Research (Railroad)										Compressor Blades						1807	1458	1769					
	1012																						
Collision Research (Ships)										Compressor Noise								99					
	435	436	847				1027			Compressors	190			1583	404	895		428	189				
														814	1685			1009					
															2075			1279					
Columns										Computer Aided Design				263		185	1536		188	2069			
	1311	1312	313	214															2068				
			1952	523	314																		
Combustion Engines										Computer Aided Techniques				1940	2031	732	1773	894	1035		1948	1789	
														2030		892	1933	1864	1465		1889		
															1772		2034				1939		
Combustion Noise										Computer Programs				150	71	122	123	174	25	16	197	18	459
	1843	204					227							250	151	172	163	354	255	36	297	138	589
Commercial Transportation														440	261	192	173	484	325	226	307	348	629
														670	281	262	383	674	365	326	337	458	679
Compacting														700	511	322	573	684	375	576	457	668	869
														870	591	562	653	864	415	676	537	678	1059
Compaction Equipment														1060	671	672	673	1214	605	846	547	798	1359
														1080	1161	702	823	1244	645	1096	587	848	1399
Complementary Energy Methods														1160	1181	812	1133	1254	675	1256	677	868	1419
														1180	1431	852	1243	1384	1105	1736	797	958	1429
														1430	1551	1022	1253	1394	1205	1796	867	1058	1549
														1460	1561	1172	1363	1554	1255	1866	1007	1108	1559
														1550	1741	1182	1483	2064	1455	1906	1017	1138	1579

Abstract

Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197

Volume 9

Issue 1 2 3 4 5 6 7 8 9 10 11 12

Computer Programs (Continued)							Continuum Mechanics	
1560 2071 1252 1503	1555 2066 1257 1258	1739					1880	273
1740 1312 1553	1556 1617 1558 1719						Contour Mapping	
1810 1332 1723	1905 1737 1738 1809							734
2070 1502 2073	2065 1797 1778 1819						Control Equipment	
2170 1552	2145 1877 1788 2069							1765
	1562 1907 1908 2089							
	2072 2067 2068 2119							
	2097 2148							
Computer Simulation							Control Systems	
		1257					use	Control Equipment
Computerized Simulation							Conveyors	
122 173 174 1395	1476 1387 1328 2029							1859
172 1393	1496 1917						Cooling Fans	
	2036						use	Fans
								and Cooling Systems
Concrete Construction							Cooling Systems	
2150 2042 1684 545	1096				799		1221	
	1095							818
Concretes							Cooling Towers	
1101		786			359		1033 964 965	
							1064	
Configuration Effects							Cornering Effects	
use	Geometric Effects							1026
Conformal Mapping							Correlation Techniques	
120					539		70	
							1524 1945	
Conical Shells							1440	
		636						
Constitutive Equations							Correspondence Principle	
710								1276
Construction Equipment							Coulomb Friction	
1851	1534 755 346						531 692 1134	
	985 1666						1762 1434	
	1195							1077 1148 2086
	1545							1577 1818
	1665							
Construction Industry							Coupled Response	
		1545					1650 1291 1292 1103 1184	
							1920 1813 1474	
							1873	
								606 1237 1178 929
								1579
								1869
Containers							Coupled Systems	
					898			666
					1698			928
Containment							Couplings	
							1633	
	533							1627
								319
Continuous Beams							Crack Detection	
								478
	2112							
Continuous Parameter Method							Cracked Media	
							431	
							1073	
	1725	147						

Abstract

Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197

Volume 9

Issue 1 2 3 4 5 6 7 8 9 10 11 12

Cranes (Hoists)										- D -							
Crankshafts										368							
820										998							
Crash Research (Aircraft)										Damage Prediction							
1500 1501 1172										799							
195 436 1757										1189							
785										Damped Structures							
2090 1271 912										1066 1897 968 429							
Crashworthiness										Dampers							
1020 1021										142							
1500										1762							
1757 1388 399										1009							
2027										1019							
1389										2189							
1450 1811 922										Damping Coefficients							
1470 1432										1364 36 657 478 69							
1704										1806 1107 1108 279							
1636 888										1307 1618 1109							
Critical Speeds										Damping Effects							
980 251 423										1060 1882 693 1404 1575 1576							
2193										1288 479 2039							
2191										1425							
Cross Correlation Technique										Damping Materials							
241 1142 23										1425							
Curve Fitting										Damping Values							
1728										1111							
Curved Beams										Dams							
2110 1										518							
1790 801										664 1185 986							
2010										1854 1186							
30										109							
Cushioning										Damping Values							
use Impact Shock and Insulation										1111							
Cutting										Data Display							
370 2182 1694 1695										459							
1200										188							
1794 1796										Data Processing							
1200										Data Reduction							
use Data Processing										Design Procedures							
1200										2034							
1541 1967										1571							
Cylinders										1701							
20 931 683 524 525 1116 47 698 919										2031							
700 1621 1313 1314 1315										1158 1799							
940 1813 1814 1815										1571							
Cylindrical Bodies										1701							
use Cylinders										2031							
130 351 772 773 534 775 126 127 328 329										1571							
770 771 1822 963 774 1335 536 327 1648 639										1701							
960 1141 1982 1333 1334 1645 776 537										1571							
1980 1711 1643 1644 1825 966 967										1571							
1981 1823 1824 2145 1646										1571							
2143 2144 1826										1571							
Diagnostic Instrumentation										900 712 713 714 895 1586 1087 1588 899							
1279										1772 1084 1935							
1929										1934							

Abstract

Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197

Volume 9

Issue	1	2	3	4	5	6	7	8	9	10	11	12
-------	---	---	---	---	---	---	---	---	---	----	----	----

Diagnostic Techniques		Drillships	
1280 1081 712 23 304 405 406 317 228 49		use	Drills
1770 1281 892 73 404 445 896 897 318 369			and Ships
1930 1761 1082 893 584 495 1086 1277 478 1079			
1940 1771 1282 1083 894 715 1436 1437 898 1279			
1931 1582 1383 1584 1085 1586 1587 1278 1589			
1941 1932 1583 1774 1585 1936 1937 1928 1769			
2101 2102 1773 1933 1934 1935 1936 1937 1938 1939			
2103			
Diagnostics (Biomechanics)	164	Drop Tests	1483
Diesel Engines			
1221 203 204 385 1216 387 388 1239		Ducts	
384 1015 1867		30 101 932 933 354 145 526 97 98 99	
		100 761 1122 1123 934 975 1316 527 588 589	
		760 1121 1622 1623 1124 1816 757 758 759	
		1120 1321 1962 1624 1117 1118 1119	
		1320 1461 2062 1317 1318 1129	
		1460 1961 1817 1319	
		2120 1509	
		1779	
		2119	
Difference Equations	1238		
Differential Equations		Duffing's Differential Equation	
650		1040	1416
Digital Simulation		Dynamic Analysis	
210 632 453 556 1889		560 1125 437 559	
460 2122 2036		1829	
890		Dynamic Antiresonant Vibration Isolator (DAVI)	
		808	
Digital Techniques		Dynamic Balancing	73
1540 1932 1933 1934 1935 1386			
Dimensional Analysis	1115	Dynamic Buckling	
		130 1431 762 743	
Disks		320 1471	
use Disks		Dynamic Excitation	
Disks		1831 2125	
2020 141 1323 1977 2178 339 929 1709		Dynamic Loads	
		use Dynamic Excitation	
Domes		Dynamic Modulus of Elasticity	
1470 1136		523	
Donnell Theory	536	Dynamic Plasticity	
		1050 281 282	
		1800	
Doors	1502 133	Dynamic Properties	
		2130 385	
Drawbars	1404 208		1625
			1805
Drills	2043 1667 1349	Dynamic Response	
		830 121 332 103 114 315 396 37 588 89	
		1030 521 1632 113 124 1055 436 507 1648 589	
		1160 751 1952 193 1044 1295 586 607 2098 959	

Abstract

Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197

Volume 9

Issue	1	2	3	4	5	6	7	8	9	10	11	12
-------	---	---	---	---	---	---	---	---	---	----	----	----

Dynamic Response (Continued)								Earthquake Damage							
921	623	2064	1525	676	627		1639	1180	2041	2172		36	1918	1919	
1031	763	2114	1955	836	677		2129		2081				2008		
1131	943			926	937										
	1173			1036	1127										
	1723			1186	1197										
	1983			1426	1457										
				1706	1927										
				1756	2127										
				2036											
Dynamic Shear Modulus								Earthquake Resistant Design							
	493							use Earthquake Resistant Structures							
Dynamic Stability								Earthquake Resistant Structures							
151	72	993		335		67	1668	1360	1181	172	173	174	175	357	198
1381	1402	2123				377		1680	1311	1102			765	1417	1268
1581		2173				987							825		1679
						2117									
Dynamic Stiffness								Earthquake Response							
90						1927		2170	701	1752	983	2084	1185	796	1177
								2151	2082				2185	1267	1178
									2171						1059
Dynamic Structural Analysis								Earthquakes							
1430	1551	852	163		1905	6	197	331	492	563	644	35	516	1917	
1970		1722				16	1957	1752	1752	983	2084	1185	796	1177	
		1882				1536	2067	1718	1718	2084	1185	2185			1179
						1726		2151	2082	2171					
						1826									1359
						2046									2149
Dynamic Structural Response								Eigenvalue Problems							
	use Dynamic Response							450	1561	452	3	234	115	236	277
								670	1881			254	235		
								1260				334	1415		
								1880							1049
Dynamic Synthesis								Eigenvalues							
							178	331	492	563	644	35	516	1917	
								use Eigenvalue Problems							
Dynamic Systems								Elastic Foundations							
2102				1566		1048	2049	80	81	82	83	844	935		797
						1238		120	121	142	313				748
								430		2112	1763				1649
								1650							
Dynamic Tests								Elastic Media							
1800	301	132	523	1614	455	196		1070		42					
	501	592	1213	1854											
	1101	2042		2104			1808								
Dynamic Vibration Absorption (Equipment)								Elastic Properties							
1350	1371	142		1165				1111	232			1575	766		278
1480						968	1479	1131	2092				1276		
				1835											
Ears								Elastic Waves							
				65		179		1120	261	232		874	1335	956	47
								1260		682			1886	1117	258
								1320							1609
								1910							1909
Earth Handling Equipment								Elasticity Theory							
				1195				1272							277
Elastodynamic Response								Elastohydrodynamic Properties							
													303		

Abstract

Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197

Volume 9

Issue 1 2 3 4 5 6 7 8 9 10 11 12

Abstract
Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197

Volume 9

Issue 1 2 3 4 5 6 7 8 9 10 11 12

Explosion Effects		Fibers	
1570 1921	264 2085 266 267	302	
	1234 1716	1302	
Explosives		Finite Difference Techniques	
	883	123	295
External Damping			338
740 72	76 1097 1288		
Extremum Principles	9	Finite Difference Theory	
		use Finite Difference Technique	
- F -			
Failure Analysis		Finite Element Technique	
1860 221 652 1773	1085	125 281 122 123 124 595 96 337 88 89	
1081	1135	830 451 642 143 334 655 136 587 138 249	
		960 471 702 703 484 695 356 857 328 279	
		1150 551 962 743 704 705 436 937 588 379	
		1160 771 1502 913 754 1155 856 1017 598 429	
		1740 1151 1522 923 884 1405 1316 2097 638 459	
		2020 1161 1243 1244 1456 768 589	
		2140 1721 1353 1804 958 679	
		2190 1741 1753 1964 1358 929	
		2031 1803 1558 1049	
		2051 1983 1718 1169	
		2121 2033 1888 1229	
			2098 1309
			1329
Fan Blades			1449
use Fans			1819
Fans			1829
220 191 162 1003 1004 95 1426 1377 818 199			2039
820 291 462 1203 1204 145			
1510 581 522 1583 1704 895			
1530 1001 1002 1703 1005			
1810 1221 1442 1863 1205			
1351 1702 2023			
1861 1842			
2021 1862		Flexibility Methods	
1942		1060	1404
2022			
Fast Fourier Transform		Flexible Couplings	
1080 1902	1794 1935 1936	1930 1322	1464
	668		1634
	1938		
Fast Fourier Transformation		Flexible Foundation	
use Fast Fourier Transform			1306 298
Fatigue (Materials)		Flexible Rotors	
	274 486		1708
Fatigue Life		Flexural Response	
890	1213 1614 485 486 487 1578		1474
	1657		
Fatigue Strength		Flexural Vibrations	
use Fatigue Life		1290 781 112 693 914 1075 86 117 2138 339	
		1820 952 1103 1154 766 1977 779	
		1152 1293 1294 1976	
		1474	
Fatigue Tests		Flexural Waves	
1112	1026	733	
	488 889		
Fiber Composites		Flight Simulation	
40 691 1202	1074 325		498 499
140	1954 1075		508
690			

Abstract

Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197

Volume 9

Issue	1	2	3	4	5	6	7	8	9	10	11	12
-------	---	---	---	---	---	---	---	---	---	----	----	----

Flight Tests		Flutter (Continued)	
	1794	1786 1787 1778	
		1796 1788	
		1856	
Flight Vehicles			
	1353		
	1793		
Floating Bodies		Flywheels	
use	Floating Structures		1872
Floating Structures		Foams	
830	954		542
930			572
			1482
Floors		Follower Forces	
861 132	64 575 786 2037	751 2142	1445
	574 785		
	784		
Flow-Induced Excitation		Footings	
use	Fluid-Induced Excitation		2094
Flow-Induced Vibration		Forced Vibration	
use	Fluid-Induced Excitation	340 941 2 293 84 5 1146	928 1339
		1040 1651 1272 1144 245 1416	1078 1849
		2020 2051 1712 1264 1265	1098 2109
		2090 1832	1138
Fluid Drives			1148
	1006		1188
Fluid-Filled Containers			1358
280 331 962 43 774 295	1327 1138		1508
961 1642 773 1274 1135		Forcing Function	
1273 1334			233
1753		Forging	
Fluid-Film Bearings			1507
	1288	Fossil Power Plants	
Fluid-Induced Excitation			1033
940 781 532 93 94 525 46 127 108 769		Foundations	
1640 891 942 943 694 535 696 327 938 919		491 802 1363 824 1685 996 1187 48 989	
941 1802 963 814 555 776 697 1158 1319		2011 1852 2024	
981 1972 1613 944 695 816 1467 1968 1459		2094	
1331 1982 1623 1214 775 956 1847 1469		Fourier Analysis	
1621 2062 1653 1274 945 1646 1967 1669		1670 1901 494	1239
2131 2132 1813 1334 1375 1866 2137 1709		Fourier Series	
1973 1764 1435 1966	1969		114
2133 1814 1815 2146			1884
1965			598
Fluid-Induced Vibrations		Fourier Techniques	
use	Fluid-Induced Excitation	use	Fourier Analysis
Flutter			
150 311 312 553 74 165 556 107 148 149		Fourier Transformation	
980 771 432 1763 554 555 1236 307 298 309		1 494	1386
1270 1341 742 1793 974 855 1786 557 1418 679			1729
1310 1671 1552 1963 1134 1135 1796 737 1458 1499			

Abstract

Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197

Volume 9

Issue	1	2	3	4	5	6	7	8	9	10	11	12
-------	---	---	---	---	---	---	---	---	---	----	----	----

Fourier Transforms
use Fourier Transformation

Fracture Properties
281 122 124
282

Framed Structures
1922 763 1044 1125 1126 1157 1359

Frames
1161 214
674 136 1607 518

Free Vibration
941 2 333 125 1346 77 328 429
961 1272 1456 957 338 939
1271 2138 949 1309
1979 2109

Freight Cars
1210 1211 193 1208 1209

Frequency
239

Frequency Analyzers
1080 1934 1935

Frequency Domain
1729

Frequency Equation
272 373 204 1276
2071 274
1014

Frequency Response
440 1 373 204
2071 274
1014

Frequency Synthesis
865 866

Friction Bearings
1620 1618 1619

Fuel Tanks
1390

Fundamental Frequency
1100 1145 1336

Fundamental Mode
2141 669

- G -

Galerkin Method
120 1883 1134 449
1964 539
1339

Gas Bearings
922 923 2115 1108
1958

Gas Turbine Blades
2035 1937

Gas Turbine Engines
1206

Gas Turbines
712 63 374 2035
373
813

Gear Boxes
721 1573 1585 897

Gear Drives
2070 528

Gears
1701 102 103 104 315 896 1127 369
1462 1463 2124 1045 1626 1627 1929
2122 2123 2125

Geometric Effects
671 672 1804 967 48
791 1844
1141

Geometric Imperfection Effects
1141 967 118

Girders
1475

Granular Materials
1859

Graphic Methods
1550 1741 422 353 459

Grids (Beam Grids)
75 337

Grinding (Material Removal)
1372 1508

Ground Effect Machines
861 973 1225 1706 377 1868 839

Abstract
Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197

Volume 9

Issue 1 2 3 4 5 6 7 8 9 10 11 12

Ground Motion		Harmonic Analysis	
360 171 472	264	1631 682	748
	494		
Ground Shock		Harmonic Balance Method	
1920	1234	1041 1042	
Ground Vehicles		Harmonic Excitation	
1440 1401 1192 1933 2034	836 1397 318	1612	135 276 477
1490	1256 1707 1888		745 1576
Ground Vibration		Harmonic Response	
170	1567 1568 709	111	
Group Velocity	47	Harmonic Waves	42
		H-Beams	
Guard Rails	208	891	
473			
1023		Head (Anatomy)	
		961	363 364 2015
Guideways	1013		128
		Heat Exchangers	
		941 1802 1473	1965 696 1967
Gun Barrels	2086		1966
			2136
Gunfire Effects		Helical Springs	
475	1758	1344	
	1938	1474	
Gyroscopes		Helicopter Blades	
254		use Rotary Wings	
284			
Gyroscopic Effects		Helicopter Noise	
72		361 2012 2013 994 25 176 27	59
		1855 566 807	
<hr/>	<hr/>	Helicopter Rotors	
		480 1793 2174	177 808
Half-Space		1310 2173	807 1408
232	7 278	Helicopter Seats	
2092		1364	1189
Hammers		Helicopter Vibration	
1373	985	1504	
Handbooks		Helicopter Vibration Effects	
use Manuals+Handbooks		1504	569
Harbors		Helicopters	
850		991 152 803 804 805 806 567 178 809	
Hardened Installations		1811 312 993 1784 1555 1556 568 869	
1551	699	362 1503 2164 2175 1686 808 1189	
Hardened Structures		992	1856 1188 1839
use Hardened Installations			1879
		Helmets	
			979

Abstract

Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197

Volume 9

Issue	1	2	3	4	5	6	7	8	9	10	11	12
--------------	---	---	---	---	---	---	---	---	---	----	----	----

Helmholtz Resonators		Human Organs
200	1779	use Organs (Biological)
High Frequency Resonance Technique	715	Human Response
		570 2001 212 1743 1504 205 26 1517 1368 569
		1190 252 1913 1744 995 206 1687 1518 1369
High Speed Transportation	1381	1230 1512 2014 1745 1016 2087 1688 1689
		1400 2034 1746 2078
		2160 2001 2076
Hitches		Human Tolerance
use Drawbars		1857
Hoists	367	Hunting Motion
		1210 1211 1212
Hole-Containing Media		2091
1481 932 933 764 1645	758	Hydraulic Dampers
1822 1124		1162
1644		
Holographic Techniques		Hydraulic Equipment
1151	66 737 958 1799	1660 344 46 1737 1738 1739
1281	1456	1376
	1976	1736
Holonomic Systems	854	Hydraulic Systems
		1700
Holzer Method	1039	Hydraulic Valves
		2137
Honeycomb Laminates	1124	Hydrodynamic Excitation
		305 2047 179
Honeycomb Structures		1409
		2099
Hopkinson Bar Technique	1642	Hydroelectric Power Plants
	196	2080 801
Household Appliances		Hydrofoil Craft
902		434
Housings	1556	Hydrostatic Bearings
		1805 1617
Hovercraft		Hydrostatic Drives
use Ground Effect Machines		1723
Human Factors Engineering		Hyperbolic Parabolic Shells
391	994	964 965 1139
Human Hand	1028	Hysteretic Behavior
		85
		Hysteretic Damping
1355 1356 1357		1751 653
2176		1981 276
Human Head		1926 1268 1229
use Head (Anatomy)		

Abstract

Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197

Volume 9

Issue	1	2	3	4	5	6	7	8	9	10	11	12
-------	---	---	---	---	---	---	---	---	---	----	----	----

- 1 -												Instrumentation			
Impact Dampers use Shock Absorbers															
Impact Noise												2016			
Impact Pairs															
Impact Response (Mechanical)															
81	522		1604		476	297	958	1149				2016			
1571	1522							1439				375		1208	1209
	1922							1809				675			1379
Impact Shock															
640		363	214		636	1747						2016			
		913			1756							2185			
Impact Testing use Impact Tests															
Impact Tests												2016			
1202		784	1095	506			979		1420	941	2063	125	1296	1297	1298 1559
1392									1421			535		1558	1899
Impedance															
1451			705				1509		2016				1435		1828
Induction Motors															
							729		2016				1895		
Industrial Facilities															
971		183	184	1565	546	347	28	1689	2016				263		1185
1891		1543	734	1665	876			1949					573		2095
			1544		2076				2016				491		856
			1564						2016						
Industrial Noise use Industrial Facilities and Noise Generation															
Inertia Relief Method												2016			
1922													721		1799
Inertial Forces															
1060	2191								2016				835		838
Inflatable Structures															
		1364		1647					2016				838		
Influence Coefficient Matrix use Influence Coefficient Method															
Influence Coefficient Method												2016			
410	222				407		409		1281			2016			
									1931			818			
Initial Deformation Effects															
1152	953			116	967				2016				Internal Damping		
									1780	2141	72	85	76	1097	1229
									2192			1075			
Internal Pressure															
												2016			
Abstract															
Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197															
Volume 9															
Issue	1	2	3	4	5	6	7	8	9	10	11	2016			

Internal Resonance		Lamb Waves										
	1034		7									
	1264											
Isolation		Laminates										
	363	use	Layered Materials									
	576											
Isolators		Landing										
180	573	640	355 636 157									
530			1756									
Isoparametric Elements		Landing Fields										
	1888	use	Aircraft Landing Areas									
Isotropy		Landing Gear										
	258	1502	548 2069									
		2002	2068									
Iteration		Landing Impact										
	1645	use	Landing and Impact Shock									
	448 649											
<hr/>												
- J -												
Jet Aircraft		Landing Pads										
	1842		1756									
Jet Engines		Landing Shock										
791	1673	use	Landing and Impact Shock									
1841												
756	1996	1996										
	598											
218	219											
Jet Noise		Landing Simulation										
460 461 722	1674	use	Landing and Simulation									
791 1262	1996											
1071 1842		1168 199										
	2158	2158										
	549	549										
Joint Stiffness		Laplace Transformation										
	105	121 1953	1065									
Joints (Junctions)			1609									
1600	316		1729									
		1820										
			1126									
		481										
	488		1878									
	1128											
	1818											
Journal Bearings		Large Amplitudes										
90 91 922 923 424 305	1107 1108 1109	1323 1184 1225	1178 1299									
920 921 1452 1453 1284 425												
		621	2194									
1110 1111 1454 1455	1449											
1450 1451	1805											
<hr/>												
- K -												
Kinematics		Lawn Mowers										
	1632		1705 927									
Lagging		Layered Materials										
	1484	680 111 112 113 2094 95 476 127 78 949										
		950 931 912 233 1145										
		1760 1341 1852 913										
		1830 1761 1923										
Abstract												
Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197												
Volume 9												
Issue	1	2	3	4	5	6	7	8	9	10	11	12

Least Squares Method		Lumped Parameter Method	
1750	657	590	704 245 1096 1177 218 1049
		1880	1447 2079
			2129
Liapunov's Method		Lyapunov Functions	
use Lyapunov Functions		2142	
Limit Analysis	2028		
Linear Analysis			- M -
use Linear Theories			
Linear Programming	223	Machine Diagnostics	
		use Diagnostic Techniques	
Linear Systems		Machine Elements	
11 1242	655	use Machinery Components	
851 2052	1885	Machine Foundations	
2082	2079	2011 1853 1685 2106	
Linear Theories		Machine Noise	
420	356	use Machinery Noise	
Linkages		Machine Tools	
1130 1632	1894 935 2126 2127 2128 1129	1000 1201 2182 1373	185 186 187 1128 999
		1160 2181	1806 577 1889 2179
Liquid Filled Containers		2070	1986 1617
use Fluid Filled Containers		2100	
		2180	
Liquid Propellant Rocket Engines		Machinery	
2040		430 1161 892	406 877 49
		1371	369
Liquid Propellants	625	Machinery Components	
		1161 1524 1636	
Liquids	572	Machinery Foundations	
		use Machine Foundations	
Locks (Waterways)		Machinery Noise	
360		971 972 1093 1094 1585 1086 577 1198 999	
Locomotives	2025	1092 1283 1664	1688 1199
		1663 2154	1699
		2153	
Longitudinal Response		Machinery Vibration	
210	1446 1857 638	1350 1941 1083 1524 1585 1086 2017	
Loves Shell Theory		1690 1584	
	1799	2101	
Lubrication		Machining	
1110 1111 1452 433 924 305 386 1507 1618 1109	810		
1453 1454 925			
1455			
Lumped Mass Method		Magnetic Tapes	1955
use Lumped Parameter Method			
Manifolds			
			597 948

Abstract
 Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197

Volume 9

Issue	1	2	3	4	5	6	7	8	9	10	11	12
-------	---	---	---	---	---	---	---	---	---	----	----	----

Manuals and Handbooks		Maxwell Fluid		
1991				128
Marine Propellers		Measurement Instruments		
	use	Measuring Instruments		
2024				
Masonry		Measurement Techniques		
2172		70 361 292 513 904 65 286 387 308 69		
Mass Half-Space Systems		1190 371 512 603 1094 385 1056 577 1198 289		
		1440 1441 1092 733 1594 395		
1265		1530 1722 1063 1704 515		
		1760 1093 1945		
Mass-Beam Systems		1950 1443		
1873	1445	1878	1743	
				1949
Mass-Spring Systems				2139
530 81 522 333	1265	529		
1873				
Material Damping		Measuring Instrumentation		
1780	1424	1076 2107	1433 1754 1945 1286	
	1924		1943 1944 2105	
Materials Handling Equipment				
1692 1693				
Mathematical Modeling		Measuring Instruments		
use Mathematical Models		1190 1441 292 603 504 385 286 287 288 1439		
Mathematical Models		1722 903 904 905 1056 1597 1438		
190 261 382 163 314 165 86 147 128 29		1433 1754 1945 1286		
200 351 412 183 754 365 646 467 188 179		1943 1944 2105		
260 411 422 193 1314 875 656 607 218 379				
480 551 452 273 1394 885 976 647 378 489				
490 601 522 573 2054 1105 1096 857 468 629				
590 961 532 583 1245 1106 1017 628 879				
600 1031 622 1013 1355 1376 1177 698 969				
800 1061 642 1353 1395 1476 1417 958 1049				
880 1451 692 1423 1695 1696 2007 998 1069				
1030 1012 1483 1845 1706 1058 1369				
1050 1252 1953 1965 1756 1068 1399				
1380 1352 2033 2025 1846 1108 1649				
1520 1422 2053 2165 2176 1258 1819				
1570 1832 2175 1508 1869				
1580 2182 1628 1879				
1610 1898 1889				
1680 2039				
1740				
1750				
1920				
2080				
Mathematical Programming		Mechanical Admittance		1878
	8			
Matrix Methods		Mechanical Elements		
1290 2072	749	1130 531		1129
				1939
Maximum Response		Mechanical Filters		
1860		1		
		Mechanical Impedance		
		364 495 2176 857 728		
Abstract		Mechanical Reliability		
Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197		use Reliability		
Volume 9		Mechanical Systems		
		1860 255 1196 1197		
Issue	1	Membranes		
	2	240 1154 245 1326 937 938		
	3	1560 1325 1466 1298		
	4			
	5			
	6			
	7			
	8			
	9			
	10			
	11			
	12			

Method of Characteristics	1609	Modal Tests	502	1413	454	455	726	727	848	1029
			1412		724	725	2106	1287	1948	
Method of Steepest Descent		Mode Shapes	310	322	423	414	245	326	1007	598
use Steepest Descent Method			990	612	1033	1644	1605	756	1157	639
Method of Superposition	957		1810	1332	1363	1974	1645		1427	2019
			2140				2145		1877	
Military Facilities									2147	
442		Model Testing	270	1401		93	94	735	266	
Military Vehicles						594		2156	268	269
400	1697	Model Tests	use	Model Testing						
Milling (Machining)	188	Modular Approach							1017	
Mines (Excavations)										
170 1781										
Minimum Weight Design	1044	745	137	2048	149	Modulus of Elasticity			106	
						1760				
Missile Components						Moire Effects				909
440										
Missile Launchers						Monte Carlo Method			1607	169
700						131				
Missiles						Moorings			2036	
500 621		325		498	619					
620				508						
				618		Motorcycles				
						841	842	1403	1465	
Modal Analysis										
420 71 412 503 604 915 666 17 218 419						Motor Vehicle Noise				
610 101 1902 593 724 1575 916 667 418 669						402		1224	215	216
960 701 1413 954						662		285		217
1300 821 1903 1134										878
1900 1901										1218
1981										
Modal Control Technique						Motor Vehicles				
						1250	501	1402	373	374
							1251			
									396	
									318	599
Modal Damping						Motors				
701	613	614	275	616		1729				
				615	1986					
				2196						
									1085	
Modal Models						Mountings				
251						1690	632			
									1505	
Modal Superposition Method						Moving Block Technique				
864							1793			
									1856	
Modal Synthesis						Moving Loads				
261 2072 253						740	772	83	795	1466
611						750	2092	753	1845	1846
						1300		1603	1955	748
						2110		1873		918

Abstract

Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197

Volume 9

Issue	1	2	3	4	5	6	7	8	9	10	11	12
-------	---	---	---	---	---	---	---	---	---	----	----	----

Moving Strips										Noise Generation (Continued)										
1955										200	1511	462	933	734	1515	1216	1517	1248		
Mufflers										1220	1531	1512	1203	1004	1545	1516		1518		
200	451		143		146		969		1640	1841	1942	1373	1064	1705			2128			
2061									1700	1861	2022	1493	1204	2075						
Multidegree of Freedom Systems																				
	1264	1885	686	1447	248															
			1566																	
Multistory Buildings										Noise Measurement										
1680		1102	1833	1834		686	797	358	169	390	371	32	543	24	205	156	27	158	159	
							1177	798	799	840	1441	292	833	64	515	206	227	1198	1199	
							1677	1358	1179	1010		862	1063	504	835		387	1398	1689	
										1230		902	1093	524	1995		577	1438	1949	
										1440		1092	1543	734			1667	1858	2159	
												1359		1442	1863	1094		1998		
												1789		1912	1993	1734		2108		
												1849		2012	2013	1994				
Musical Instruments										Noise Meters										
	231						257			use		Sound Level Meters								
<hr/>																				
- N -																				
Nacelles										Noise Path Diagnostics										
1671			1704														23			
Narrow-Band Excitation										Noise Prediction										
			244							840	441	352	353	354	25		547	548	879	
NASTRAN (Computer Programs)										880	1911	1062	1843	464	1385		817		1879	
1560	1561	1552	1553	1554	1555	1556	1557	638	1559	1520		1992								
		1562						1558		2160										
Natural Frequencies/cy										Noise Propagation										
120	91	322	773	104	245	16	277	88	79	581							878			
310	121	612	1033	414	955	326	657	598	539	60	191	62	153	54	145	176	57	368	99	
430	541	1332	1043	484	1605	336	777	778	639	260	671	372	323	144	155	216	177	758	159	
540	741	1802	1293	534	1645	756	1007	1338	849	1170	761	402	353	154	185	346	207	788	189	
650	911	1822	1323	1364	1685	1446	1037	1448	989	1510	791	672	573	184	345	546	347	988	199	
990	1301	1832	1363	1444	2135	1966	1157	1628	1299	1520	831	792	663	344	665	1216	367	1218	259	
1340	1611			1644	2145	2196	1877	1638	1779	2000	1001	832	813	354	815	1316	497	1488	389	
1810	1721				1974		1977	2148		2120	1051	842	843	544	975	1396	767	1988	789	
2020	1821									1221	932	1123	734	1015	1486	837	2018	1469		
2140	1871									1351	972	1233	794	1495	1496	877		1489		
	2091									1361	1002	1283	934	1545	1616	927		1519		
Noise Barriers										1491	1052	1403	1484	1665	1626	1217		1539		
970	971			543	144	545	1486	1487	29	1661	1462	1463	1494	1675	1666	1247		1699		
	1991					2155	2156	1987		1691	1482	1623	1544	1855	1996	1397		1989		
							2077			1701	1542	1663	1564	2155	2116	1487		1999		
Noise Control										1891	1702	1693	1624	1485	2076	1667				
										2001	1842	1893	1664	1565			1817			
										2021	1892	2023	1714				1837			
										2061	1962	2153	1874				1867			
																	1997			
																	2154		2037	
Noise Reduction																		2157		
20	581	102	183	184	1005	846	157	28	819								2194		2197	
100	661	202	203	204	1115	876	187	818	1219											
170	1011	352	213	324	1195	1116	1167	948	1509											
<hr/>																				
Abstract										Volume 9										
Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197										Issue										
1										2										
3										4										
5										6										
7										8										
9										10										
11										12										

Noise Source Identification								Nuclear Explosions										
291	182			215		877	908	189	510	511			494					
352						1217		999										
722						1377		1589										
1222						1867												
1262																		
Noise Tolerance								Nuclear Fuel Elements										
371				2014	475	566	1687	2078	590	2011	822	593	124	585	596	197	198	
									830		1382	823	194	705	706	827	768	
									1560		1522	1363	704	785	826	1967	1919	
Nondestructive Tests								Nuclear Power Plants										
1772	1923					898	39		1383	824	825			2097				
Nondestructive Testing								Nuclear Powered Ships										
	use								435	436								
Nonholonomic Systems								Nuclear Reactor Components										
	854							591	592	1313	1214	2145			588	589		
								821									829	
Nonlinear Analysis								Nuclear Reactor Containment										
	use							380	381	382	383				586	587	588	
Nonlinear Damping									591	592					2047		379	
	1412										1982						589	
Nonlinear Programming								Nuclear Reactors										
	150							812			594	195	196	197				
									595	1866							707	
Nonlinear Response								Nuclear Weapons Effects										
	1761		243	864	595	296	237	238			1754	1755						
			1244															
Nonlinear Systems								Numerical Analysis										
	10	881	2102		2044	5	6	1727		1099	1730	11	123	1344	75	2046	1417	319
	1860					655					1471					2047	429	
						2045									2127	1239		
Nonlinear Theories								Oceans										
	1501					126											279	
						1046												
						1886												
Normal Modes								Off-Highway Vehicles										
	960		44			16	7		1169							1697	899	
			684			806	47											
						427												
Nozzles								Off-Shore Structures										
	202										1315						299	
																1699		
Nuclear Explosion Damage								Optical Measuring Instruments										
	642						139								906	1437		
															1946			
Nuclear Explosion Detection								Optimization										
	1072							1240	1241	12	893	424	855	76	997	608	1549	
								1831	882	2033	654	1045	576	2027	968			
Nuclear Explosion Effects									2152			1145	836					
	471	472	673	574	575		637		699			1475	1536					
			1182		674				1569									

Abstract

Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197

Volume 9

Issue	1	2	3	4	5	6	7	8	9	10	11	12
-------	---	---	---	---	---	---	---	---	---	----	----	----

Optimum Control Theory							Passenger Vehicles													
1890 571							2049	1192			835	1857								
Optimum Design							Pasternak Foundations													
180 1043 1044							1197	1301												
1890							Pattern Recognition Techniques													
Orthotropism							1940				1939									
540 1981							115 126													
1820							955 1146				397									
Oscillation							Pavement Roughness													
							790				397									
							Pavements													
							211	1684 355			359									
Oscillators							Pendulums													
651 652 13 244							1066 247 1748	1433			1656									
							653	1763												
Overhead Cranes							Periodic Excitation													
							1475	1410 1653 465			1537 1978 1099									
								1823			2079									
Overhead Guideways							Periodic Response													
1300							330 2121 772	1313 465 296			237 238 1099									
							1380	1452 306 1537			1958 1159									
Overspeed Testing								2032 1226 1727												
							718	1276 1747												
<hr/>																				
- P -																				
Packaging							Periodic Structures													
580							1574	1065												
Packaging Materials							1193	Perturbation Theory												
							1195	1140 61 242 13 624			1046 1097 2118									
								1710 421 262 243 1144			1136 1537									
Panels							320 1492	2050 1961 452												
							1270 1552	952												
							674 765 766 107 108 39	Photoelastic Analysis												
							874 737 478 679	290 122 1783												
							1134 938 1429	Photographic Techniques												
							1964 1639	909												
Parachutes							1252 163	Pile Foundations												
								1324												
Parameter Identification Technique							1411 152 2053 454 455 1246 657 658 349	Pile Structures												
							1731 2052 2073 2054 2175 2177 858	80 1032			1925 2096									
							2118	1362												
Parametric Excitation							520 2123	Pipelines												
							1130	1641			525									
							335 1066	Pipes (Tubes)												
							2086	280 381 382 383			295 767 109									
								380 532 2133			787 379									
Parametric Response							1140	1640 2132			1327 769									
							2045	1077												
Abstract							Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197													
Volume 9																				
Issue	1	2	3	4	5	6	7	8	9	10	11	12								

Pipes (Tubes) (Continued)													
			1467		1129								
					1469								
Piping Systems							Polar Correlation Technique						
321	322	1033	324	945	946		1262						
1802			944		1966								
			1334				1328						
							1468						
							1968						
Pistons							Polymers						
1610			1015	386			1434						
			1705										
Plastic Deformation							Polyurethane Resins						
541	342						574						
Plastic Properties			915										
Plastics							Porous Materials						
		1533					871						
Plates							Power Plants (Facilities)						
120	81	12	93	114	95	66	use	Electric Power Plants					
540	111	112	113	334	245	476	1267						
750	121	122	693	874	335	506	1328						
780	341	132	733	914	1145	866	1468						
950	541	332	783	954	1845	956	1968						
2140	781	782	953	1144	1975	1146	1337	1148	1149				
	951	952	1143	1154		1336	1747	1538	1809				
	1151	1152	1343	1444			2147	1638	1819				
	1341	1272	1473	1974				2138	1979				
		1342	1973					2139					
		1472											
		1972											
		2062											
Pneumatic Equipment							Pressure Regulators						
1661	1662		985		2157	508	2129	440					
Pneumatic Lines							Pressure Vessels						
		943					533						
Pneumatic Springs							Prestressed Structures						
			2016				911						
Pneumatic Tires							Proceedings						
							883						
Pneumatic Valves							Probability Theory						
1970							2080	652	263				
Pogo Effect													
481													
Pogo Oscillation							Propeller Noise						
use	Pogo Effect						153						
Point Source Excitation								25					
			1975					1995					
Abstract							Propulsion Systems						
Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197	
Volume 9													
Issue	1	2	3	4	5	6	7	8	9	10	11	12	

Pulse Excitation		Random Response									
130	1273	2080	82								
			85								
Pumps		Random Vibration									
1701	812	1540	881								
1583	2183	992	1751								
		1214	1142								
		635	1196								
		686	1749								
		487	488								
		1752									
Pyrotechnic Shock Environment		Rapid Transit Railways									
883	514		376								
<hr/> - Q - <hr/>											
Quadratic Damping		Rayleigh Waves									
1163		42	33								
		483	7								
<hr/> - R - <hr/>											
Rail Transportation		Reciprocal Measurement									
1520			736								
Railroad Cars		Reciprocating Engines									
1210	1211	1212	193	584	375	676	677	1208	1209	1932	1523
			513		675	1106		1838		386	
						1436					389
Railroad Tracks		Reciprocity Principle									
				98							
		Recording Instruments									
		1781		2105							
Railroad Trains		Rectangular Membranes									
1010	1011	192	1513	1514	1515	1516	867	868	1519		
			1381		2025	2026	1517	1518		1637	939
Railroad Vehicles		Rectangular Plates									
use	Railroad Trains	1150	1821	2142	333	914	115	116		1338	779
		1650	2141		1153		955	336		1978	1339
		1820									
Railroads		Reduction Methods									
1060			778								
Rails		Re-entry Vehicles									
use	Railroad Tracks	1570		636	637						
		1720									
Railway Vehicles		Regulations									
use	Railroad Trains	660	161	372	1543	1544	1545	2076	217	1248	659
		860	661	1052	1893	2154			1247		1949
				1051	1542						
				1891	1892						
Random Decrement Technique		Reinforced Beams									
1793			1954								
Random Excitation		Reinforced Concrete									
870	453	224	135	1066	657	1468	1979				
1410			485	1566	997	1748	2079				
			995		1067			124		1347	168
			1065		1637			1834		1607	989
			1707								1639
Random Parameters		Reinforced Plastics									
			1202								
1048											

Abstract
Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197

Volume 9

Issue	1	2	3	4	5	6	7	8	9	10	11	12
-------	---	---	---	---	---	---	---	---	---	----	----	----

Reinforced Plates										Road Tests (Ride Dynamics)													
										use Ride Dynamics													
Reliability										Rocket Components													
										2040													
Resonance										Rockets													
										227													
Resonance Bar Techniques										Rods													
										1130 751 752 1303 74													
Resonance Tests										1241 1302 1313 1104													
										1601 1802 2113													
Resonant Bar Technique										1801													
use Resonant Bar Technique										Roller Bearings													
										303													
Resonant Frequency										713													
850	1342	1274	165	1427	1169	Rooms										131 64							
1710	2162	2144	2147			441																	
Resonant Response										Rotary Inertia Effects													
10 1981	1342	43	96	1237	1968	use Rotatory Inertia Effects																	
1710	2090					936																	
Resonators										Rotary Seals													
										936													
Reverberation Chambers										Rotary Wings													
1590	241	1887	719	990	1261	92	2174	176	177	1408	869												
1691				1310	1811	312		846	1457		1959												
Reviews										Rotating Structures													
550 1001	862	723	904	1055	656	857	888	709	1280 1711	254	895	406	418	339									
1420 1421	1062	853	1054	1425	896	1057	1188	1049	2020 1871			896	2178	369									
2060 2061	1422	863	1074	1495	1056	1897	1898	1419				1586		429									
	2062	903	1424	1895	1426	2058	1899						439										
	2172	1053	1734	1896	1896	2059	1981																
	1423	1894																					
	2063	2104																					
Ribs (Supports)										Rotatory Inertia Effects													
1151		938	910	111	742	743	744	225	1156	517	118	749											
			1340	1291	1292	1323		955	2146			1299											
			1830	1801	1652	2093	1981		1445														
Ride Dynamics										Rotor-Bearing Systems													
1190	2023	2034	1985	1256	2087	1399	420 411 422 1453	224	415 306	408	1229												
1400							1872	924	845 1306	428	1419												
									925 1406	1008													
									1455	1288													
Rigid Foundations										Rotor Blades (Rotary Wings)													
		797								use Rotatory Wings													
Rings										Rotor Blades (Turbomachinery)													
130	822	1653	1156	129			1730 311 992	1114	2075 2116	927	2118												
340		1963	1826							1457													
1830										2117													

Abstract

Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197

Volume 9

Issue	1	2	3	4	5	6	7	8	9	10	11	12
-------	---	---	---	---	---	---	---	---	---	----	----	----

Rotor-Induced Vibration										Secondary Waves					
										804	808	809	1275	986	1267
Rotors (Machine Elements)															
180	221	72	223	404	405	416	407	218	219						
220	251	222	413	424	425	606	417	608	409						
410	421	332	423	844	1405	1226	427	718	419						
420	431	422	893		1445	1406	607	988	429						
1530	811	432	993		2035	1586	987	1288	1529						
1710	1711	1582	2193		2175	1616	1407		1579						
1870	1871	1712							1709						
2060	1941	1932													
2190	2191	2192													
Runways															
211				355											
<hr/>															
- S -															
Safety Belts															
use	Seat Belts														
Safety Restraint Systems															
1016															
Sand															
493															
523															
Sandwich Laminates															
use	Sandwich Structures														
Sandwich Panels															
use	Panels														
	and Sandwich Structures														
Sandwich Structures															
320				114		766		39							
						1146		79							
Satellite Antennas															
use	Spacecraft Antennas														
Satellites															
611				623	614	625									
					624	635									
Saws															
1115				1147	2178										
Scaling															
1115															
Seat Belts															
1393				1016											
Seats															
403				1345											
Abstract															
Numbers:	1-231	232-447	448-647	648-850	851-1036	1037-1235	1236-1414	1415-1534	1535-1724	1725-1879	1880-2043	2044-2197			
Volume 9															
Issue	1	2	3	4	5	6	7	8	9	10	11	12			

Shells											Shock Measurement								
330 341 332 293 124											use	Measurement Techniques							
750 1421 962 923											and Shock Response								
1330 1471 1142 1333											Shock Pulse Method								
2071											1770	1082							
Shells of Revolution											Shock Resistant Design								
											123	1829	263		467	198			
											1143				468				
Ship Hulls											1291 1422 1423	225 1896	1898	Shock Response					
											1720	621	1073	904	1056	1567	1828	469	
											951	1143			1597	2048	1789		
Ship Structural Components												1781			1827				
Ship Vibration											1422 1423	1896	1028	Shock Response Spectra					
												1898		1053					
															609				
Shipboard Equipment Response											1713					739			
Shipboard Machinery											1191			Shock Tests					
														1540 461 1232 1783 594		1698	469		
												1713		1754		509			
																619			
Shipping Containers											991 1202 1193 1194	1574		Shock Tubes					
														61		1285			
Ships											1421	1713 1714	746 2037	Shock Waves					
												609		1530 1921 1303					
												619		1570					
												1409							
Shock Absorbers											1162 343	37 1348 1349		Shock Wave Propagation					
											1163			270		884 265 196 267 268 269			
														700					
														1460					
Shock Absorption											342	1475		Shrouds					
														1233		1309			
Shock Excitation											130 621	1953	1167 1069	Shuttles (Spacecraft)					
											740 1071			1233					
											1920								
Shock Isolation											572	574 575	1058	Signal Processing Techniques					
														1940		1936 1587 1938 1939			
														1933 1934		1937			
Shock Isolators											1194	996		Silencers					
														1662					
Shock Loads											use	Shock Excitation		Silicone Resins			579		
														Silos (Missile)					
														use	Missile Silos				

Abstract

Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197

Volume 9

Issue	1	2	3	4	5	6	7	8	9	10	11	12
-------	---	---	---	---	---	---	---	---	---	----	----	----

Simulation					Sonic Boom						
510	511	63		37	148		63	905	1546	457	
							643	1495		1547	
										2057	
Single Degree of Freedom Systems											
	865	1196		2088							
	2045	1576									
Skew Plates					Sonobuoys						
	1652										87
Skis					Sound Attenuation						
		647									97 1378 2119
Slabs					Sound Generation						
					1003		1124	875	66	257	1308
					1200				1816	917	1318
				1639							
Slamming					Sound Insulation						
	434				use	Acoustic Insulation					
Slider Bearings					Sound Level Meters						
	2114			1449	582	1593	1594	1595			1438
Slider Crank Mechanism					Sound Measurement						
1631	2093					903	1594		376		
Slip Joints					Sound Pressures						
1630					241					527	1719
		1577		2089	1261						
Stooshing					Sound Propagation						
	43	774		1137	618*	1622					19
Snap Through Problems					Sound Reflection						
320	1333			126							98
Snowmobiles					Sound Transmission						
	843				240	351	22	463	144	45	786
					681	782	1343	1924	345		787
Soil Compacting					1511		1513	1944	1945		98
490				489						1117	788
Soils										1317	1118
	1851									1497	1318
		1766	447	2098							1468
		1926	1267								1548
Solar Arrays					Sound Transmission Loss						
		616		629	1950		133	134		767	109
								684			969
											1599
Solid Propellant Rocket Engines					Sound Waves						
1410				738	680	691		1273	34	685	
					690	1121			694	1975	
Solid Propellant Rockets					1120	1961					1909
630					1910						
Solid Rocket Propellants					Space Stations						
				438	230						
										2038	1719
Sonar											2039
1470											

Abstract

Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197

Volume 9

Issue	1	2	3	4	5	6	7	8	9	10	11	12
-------	---	---	---	---	---	---	---	---	---	----	----	----

Abstract

Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197

Volume 9

Issue 1 2 3 4 5 6 7 8 9 10 11 12

Steering Effects				Structural Resonance			
2032		1526	1478			2005	
		2026					
Stick-Slip Response				Structural Response			
	1374		1047		643	644	645
					36		229
					1034		799
Stiffened Plates					1234		849
1151							1159
Stiffened Shells				Structural Synthesis			
		536		1370		1654	1655
Stiffened Structures							1038
1721	1243			Subharmonic Oscillations			
Stiffness				1041	1042		
1951	1362	1833	2024	1075	1956	1128	
1672				1625			
Stiffness Coefficients				Substructure Coupling			
1450	1601	922		use	Component Mode Synthesis		
1074							
1704				Successive Approximation Method			
							37
Stiffness Methods				Supersonic Aircraft			
1060	1562			1493	1494	1495	2166
2072				2163			2197
2082							1499
Stochastic Processes				Supports			
2111				2190	2		
		486		299			
		1236					
Storage				Surface Effect Machines			
270				1380		1525	1426
		266	267	268	269		
Storage Tanks							1409
331	962			Surface Roughness			
				1602	1013	1525	
						1507	1869
Strings						1707	
301	1612	753		Surges			
1611		1443		86			814
				1326			
Strips				Surveys			
				use	Reviews		
Structural Design							
1991				Suspended Structures			
2041				991	1312		
Structural Components				Suspension Bridges			
use	Structural Members			980	981	864	165
						356	1847
						2168	
						1176	
						1676	
Structural Elements				Suspension Systems (Vehicles)			
use	Structural Members			1490	401	654	1395
					571	1164	1465
					841	1394	2025
						1506	2068
Structural Members							
1240	41	1802	673	1524	1575	137	138
1740	341		1733		1755	1657	1158
1831							
				Symposia			
				use	Proceedings		

Abstract

Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197

Volume 9

Issue	1	2	3	4	5	6	7	8	9	10	11	12
-------	---	---	---	---	---	---	---	---	---	----	----	----

System Identification	Testing Techniques											
152 373 374	50 291 62 63 64 635 426 67 68 729											
1732 2083 384	510 731 562 723 304 725 726 727 308											
	630 2031 732 1793 514 735 1026 1287 738											
	730 1022 1933 724 1425 1076 1347 848											
	860 1322 1774 1176 1797 1788											
	1600 1732 1784 1796 1798											
	1790 1792 1854 1876											
	1840 2106											
<hr/> - T - <hr/>												
Tanker Ships												
1230	1027											
Tanks (Combat Vehicles)												
	Textile Looms											
	838 1031 182 183 184 917											
	1531											
	1891											
Tanks (Containers)												
1232 1753												
Taxiing Effects												
790	Thermal Excitation											
	1473 498 769											
Temperature Effects (Excitation)												
use Thermal Excitation												
Test Data												
use Experimental Data	Thread Cutting											
	1635											
Test Equipment and Instrumentation												
50 501 722 1783 64 505 426 507 1598 619	Three Dimensional Problems											
720 1782 1284	1799											
1780	506 1777											
	1776											
Test Facilities												
60 501 62 53	Timoshenko Theory											
400 901 502 503	1100 1291 1292 743 84 295 517 1098											
500 1591 862	285 496 497 498 289											
	1301											
	717 718 499											
902	1881											
1442	1347 1088 1089											
1592	1778											
1942	1798											
Test Fixtures												
use Test Facilities	Tires											
	210 1052											
	396 1218											
	1658											
Test Instrumentation												
use Test Equipment	Tools											
	1147 1488											
Test Models	1667 1858											
592	Torque											
	1931											
Testing Apparatus												
use Test Equipment and Instrumentation	Torsional Excitation											
	1324											
Testing Equipment												
use Test Equipment and Instrumentation	Torsional Response											
	1960 1951 802 1963 1114 106 998 1419											
Testing Instrumentation	1212 1184 796 1178											
use Test Equipment and Instrumentation	1474											
	2094											
Testing Machines												
use Test Equipment and Instrumentation												
Abstract												
Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197												
Volume 9												
Issue	1	2	3	4	5	6	7	8	9	10	11	12

Torsional Vibration										Transmissibility																							
720		71	1302	1103	294	1405	1057		648	1039	use		Transmissivity																				
820		2091	2122	1907		2019																											
1600																																	
2070																																	
Towed Bodies										Transportation Effects																							
use		Towed Systems								580		512	1574																				
Towed Systems										Transportation Systems																							
1803		1105		1447						use		Transportation																					
										and Transportation Vehicles																							
Towers										Transportation Vehicles																							
800		1683		984	1236		1007	299		1573		2087																					
Tracked Vehicles										Transverse Shear Deformation Effects																							
400				377		838	839	1399		1340		111	742	743	744	225	1156																
										1830		1291	1292	1323	517		118																
										1801		1652	2093	2146		1299																	
Tracking Filters										1981																							
1941										Truck Tires																							
Tractors		403						899		840		1476																					
										1220		1477																					
Traffic Induced Vibrations										Trucks																							
								709		1221		32	603	216		217	318																
Traffic Noise										402		367																					
860	31	212	1063	464	205	206	2077	1658	259	602		1217																					
1010	1061	1062	1263	1744	545	1746	879		1052		1477						1248																
1911	1572	1543	1745		1759		1912		1913		1697						1697																
Trains										Trusses																							
use		Railroad Trains								1312		1625																					
Tramways		1512								Tubes		940	941	942	1273	1824	1135	2136															
										1642		1965						1158															
										2135																							
Transducers										Tuned Dampers								2088															
1082		1754		2105	1087				1000																								
				1944																													
Transfer Matrix Method										Tunnels								266															
322	1243					857		518	1049	270		441	267		268	269																	
Transient Excitation										Turbine Blades																							
731	1643		1794			1427		1878		310		1054		1585	96	307	308	309															
										1090		1007		378	1309																		
Transient Response																		1277															
110	71	1812	583	654	1105	306	1537	1728	219			1808		1769	1437		1809																
520	591	1153		1384	1575	856	1737	1159																									
2080	851	1903		1126								Turbine Components						1384															
				1141		1953				2035		96	718																				
Transient Vibrations										Turbine Engines								1206															
				1266		678																											
Abstract										Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197																							
Volume 9										Issue		1	2	3	4	5	6	7	8	9	10	11	12										

Turbines	162	1864	895	1007	428	2019
			1865		2018	

Turbofan Engines	1862	815
		1035

Turbofans	816

Turbomachinery								
310	101	443	444	445	306	417	1598	1079
1940			894	895	426	527		1459
2050					446			1529
2060					1036			

Turbomachinery Noise	2197

Turbulence	1173	1765	108	1709
280			1298	1729

- U -

Ultrasonic Techniques	1923

Ultrasonic Tests	use	Testing Techniques

Unbalanced Mass Response	284	415
	444	

Underground Explosions	1072	494	799

Underground Structures	270	472	266	267	268	269
700			966			699

Underwater Explosions	735	1557
	2085	2047

Underwater Sound	21	44	45	286
	684	685	1916	
		1915		

Underwater Structures	1143	125	89

Urban Noise	260	259

Urban Transportation	1911

- V -

Valves	100	1971	323	324	1285	947
	190					

Van der Pol Method	4

Variable Cross Section	1150	341	1323	1294	1155	336	1317	1338	749
	1330	451					1977	1978	1299
	1980	741							1609
			1121						
			1651						
			1951						
			1961						
			2141						

Variable Material Properties	341

Variational Methods	450	451	143	295	137	249
				1307		

Vehicle Wheels	1026

Velocity	1711
	1931

Ventilation	908

Vertical Takeoff Aircraft	352	353	354

Vibrating Foundations	935	1656

Vibrating Structures	1890	1041	1042	2044	875	66	137	1748	1859
					996	917			

Vibration Absorbers	use	Vibration Absorption (Equipment)

Vibration Absorption (Equipment)	1350	141	2152	803	1164	1057
			1480			

Vibration Analyzers	712	1588

Abstract

Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197

Volume 9

Issue	1	2	3	4	5	6	7	8	9	10	11	12
--------------	---	---	---	---	---	---	---	---	---	----	----	----

Vibration Control										Vibration Meters										
1840 811 552 1283 1714 185 166 1337 318 319										1091										1947
1541 1523 1844 985 366 838										Vibration Monitoring										1937
1713 1365 946 1188										1582 2183										
1505 1086 1568										Vibration Prediction										1879
1535 1686 2178																				
2155 2136																				
Vibration Dampers										Vibration Recording										
224 1986 1057										1781										2105
1987																				
Vibration Damping										Vibration Reduction										
1641 432										use Vibration Control										
2091 482										Vibration Resonance										
1312 1206										use Natural Frequencies										
2006																				
Vibration Detectors										Vibration Response										
1090 907										80 231 602 203 4 75 166 167 18 119										
1947 1947										170 321 313 364 425 316 747 308 129										
										1150 631 603 424 565 746 977 528 439										
Vibration Effects										1640 801 893 564 585 1926 1357 1968 599										
1112 716 1867 1078 909										2060 931 953 744 1305 2176 689										
786										1151 1503 1355 749										
906										1555 1969										
										1875 2059										
Vibration Energy Method										2165										
										608										
Vibration Excitation										Vibration Signatures										
570 1054 446										1584										
1790										1774										
1910																				
Vibration Frequencies										Vibration Spectra										
956										use Vibration Response Spectra										
Vibration Isolation										Vibration Tests										
1690 201 552 2034 1616 1187 1058 189										500 1731 1202 723 724 505 726 557 68 499										
1191 1682 2016 1697 1188 1129										730 1232 1783 764 725 1456 727 168 639										
1371 2017 1519										1410 1592 1284 1175 1717 728 729										
1691										2010 1782 1574 1405 2187 1358										
Vibration Isolators										2162 2004										
1350 1192 1193 654 997 809										Vibrators (Machinery)										
1370 804 1057 1987										490 1692 164 1775 508 359										
										1684 489										
										729										
Vibration Measurement										Vibratory Conveyors										
1230 721 832 403 304 65 376 1667 288 859										use Vibrators (Machinery)										
1760 861 513 904 395 395 1056 568 1789										and Materials Handling Equipment										
1281 903 1864 1345 1086 1398										Vibratory Techniques										
1521 1083 1715 1356 1858										1851 1532 1533 164 447										
1443 2015 2035										2105 1684										
2195										Vibratory Tools										
										1534										

Abstract
Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197

Volume 9

Issue	1	2	3	4	5	6	7	8	9	10	11	12
-------	---	---	---	---	---	---	---	---	---	----	----	----

Viscoelastic Core-Containing Media	114	78	Wave Diffraction	1070
Viscoelastic Damping			Wave Equation	1318
340 481	85	78		
480 711				
770			Wave Number	
2100			use Frequency	
Viscoelastic Foundations		1298	Wave Propagation	
		40 261 42 483 234 465 1886 1267 258 1609		
		680 271 232 733		
Viscoelastic Media	283	1915	710 1961 242 1303	
710 1581		1927	1852	
1801		1768		
Viscoelastic Properties			Wave Reflection	33
912	84	1575	912	
1472	294	1276	1472	
2112			1649	
Viscous Damping			Waveguide Analysis	
930 531	1134	245	1260	1325
Vortex-Induced Vibration		1206	1649	
350	93	94	1852	
Vortex Shedding		2116	Weapons Effects	
20 981 2022 1203	165	698		1758 139
520 1621	1613		Weapons Systems	1698
Vulnerability			1571	1434
	1234		Welded Joints	1533
			1532	1533
			Welding	
			210	1436
			Wheelset	
			1212	
			Whirling	
			1110 221 422 1453 414 845 416 77	
			431 942 1454 1455	1447
			2101 1872	
Wankel Engines			Wind-Induced Excitation	
201			800 901 992 1183 314 55 1176 1847 1848 169	
Washing Machines			2041 1312 1613 964 165 1676 2007 2168 229	
181			1402 1683 1174 965 2006	
Water		636	1184	849
640	22			1579
Water Pipelines				
		787	Wind Tunnel Tests	
			620 561 2073 1794 1785 1786 307 1798	
			800 1671 2134 1795 2006 1997	
Water Waves			1670 1791 2164	2167
901		1315	1840	
			1870	

Abstract

Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197

Volume 9

Issue	1	2	3	4	5	6	7	8	9	10	11	12
-------	---	---	---	---	---	---	---	---	---	----	----	----

Wind Tunnels	51	52	53	54	55	56	1088
Wind Turbines							1007
Windmills							984
Wing Stores	362	977 1498 1169					
	562						
Winkler Foundations	83	126 2109					
Wire	1304	1886					
Woodworking Machines							789

Abstract
 Numbers: 1-231 232-447 448-647 648-850 851-1036 1037-1235 1236-1414 1415-1534 1535-1724 1725-1879 1880-2043 2044-2197

Volume 9

Issue	1	2	3	4	5	6	7	8	9	10	11	12
-------	---	---	---	---	---	---	---	---	---	----	----	----

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AERONAUTICAL QUARTERLY Royal Aeronautical Society 4 Hamilton Place London W1V 0BQ, UK	Aeronaut. Quart.	BULLETIN OF JAPAN SOCIETY OF MECHANICAL ENGINEERS Japan Society of Mechanical Engineers Sanshin Hokusei Bldg. H-9 Yoyogi 2-chome Shibuya-ku Tokyo 151, Japan	Bull. JSME
AIAA JOURNAL American Institute of Aeronautics and Astronautics 1290 Ave. Americas New York, NY 10019	AIAA J.	BULLETIN OF SEISMOLOGICAL SOCIETY OF AMERICA Bruce A. Bolt Box 826, Berkeley, CA 94705	Bull. Seismol. Soc. Amer.
APPLIED MATHEMATICAL MODELING IPC House, 32 High Street Guildford Surrey GU1 3EW, UK	Appl. Math. Modeling	CIVIL ENGINEERING (NEW YORK) ASCE Publications Office 345 E. 47th St., United Engr. Ctr. New York, NY 10017	Civ. Engr. (N.Y.)
ARCHIVE FOR RATIONAL MECHANICS AND ANALYSIS Springer-Verlag New York Inc. 175 Fifth Ave. New York, NY 10010	Archive Rational Mech. Anal.	CLOSED LOOP MTS Systems Corp. P. O. Box 24012 Minneapolis, MN 55424	Closed Loop
ARCHIVES OF MECHANICS (ARCHIWUM MECHANIKI STOSOWANEJ) Export and Import Enterprise Ruch UL Wronia 23 Warsaw, Poland	Arc. Mech. Strosowanej	COMPUTERS AND STRUCTURES Pergamon Press Inc. Maxwell House, Fairview Park Elmsford, NY 10523	Computers and Struc.
ATM MESSTECHNISCHE PRAXIS R. Oldenbourg Verlag GmbH Rosenheimer Str. 145 8 München 80, W. Germany	Messtech- nishe Praxis	DESIGN NEWS Cahners Publishing Co., Inc. 221 Columbus Ave. Boston, MA 02116	Design News
AUTOMOBILTECHNISCHE ZEITSCHRIFT Franckh'sche Verlagshandlung Abteilung Technik 7000 Stuttgart 1, Pfizerstrasse 5-7 W. Germany	Automobil- tech. Z.	DIESEL AND GAS TURBINE PROGRESS Diesel Engines, Inc. P. O. Box 7406 Milwaukee, WI 53213	Diesel and Gas Turbine Progress
AUTOMOTIVE ENGINEER P. O. Box 24, Northgate Ave. Bury St. Edmunds Suffolk IP32 GBW, UK	Auto. Engr.	ENGINEERING MATERIALS AND DESIGN IPC Industrial Press Ltd. 33-40 Bowling Green Lane London EC1R, UK	Engr. Matl. Des.
BALL BEARING JOURNAL (English Edition) SKF (U.K.) Ltd. Luton Bedfordshire LU3 1JF, UK	Ball Bearing J.	ENVIRONMENTAL QUARTERLY Environmental Publications, Inc. 252-46 Leeds Rd. Little Neck, NY 11362	Environ. Quart.
BAUINGENIEUR S. Hirzel Verlag, Postfach 347 D-700 Stuttgart 1, W. Germany	Bauingen- ieur		

PUBLICATION AND ADDRESS	ABBREVIATION	PUBLICATION AND ADDRESS	ABBREVIATION
EXPERIMENTAL MECHANICS Society for Experimental Stress Analysis 21 Bridge Sq., P.O. Box 277 Westport, CT 06880	Exptl. Mech.	INTERNATIONAL JOURNAL OF ENGINEERING SCIENCES Pergamon Press Inc. Maxwell House, Fairview Park Elmsford, NY 10523	Intl. J. Engr. Sci.
FORSCHUNG IM INGENIEURWESEN Verein Deutscher Ingenieur, GmbH Postfach 1139, Graf-Recke Str. 84 4 Dusseldorf 1, W. Germany	Forsch. Ingenieurw.	INTERNATIONAL JOURNAL OF MACHINE TOOL DESIGN AND RESEARCH Pergamon Press, Inc. Maxwell House, Fairview Park Elmsford, NY 10523	Intl. J. Mach. Tool Des. Res.
GAS TURBINE INTERNATIONAL Gas Turbine 80 Lincoln Ave. Stamford, CT 06904	Gas Turbine Intl.	INTERNATIONAL JOURNAL OF MECHANICAL SCIENCES Pergamon Press, Inc. Maxwell House, Fairview Park Elmsford, NY 10523	Intl. J. Mech. Sci.
HIGH-SPEED GROUND TRANSPORTATION JOURNAL Planning Transportation Assoc., Inc. P. O. Box 4824, Duke Station Durham, NC 27706	High-Speed Ground Transp. J.	INTERNATIONAL JOURNAL OF NONLINEAR MECHANICS Pergamon Press, Inc. Maxwell House, Fairview Park Elmsford, NY 10523	Intl. J. Nonlinear Mech.
HYDROCARBON PROCESSING Gulf Publishing Co. Box 2608 Houston, TX 77001	Hydrocarbon Processing	INTERNATIONAL JOURNAL FOR NUMERICAL METHODS IN ENGINEERING John Wiley and Sons, Ltd. 605 Third Ave. New York, NY 10016	Intl. J. Numer. Methods Engr.
IBM JOURNAL OF RESEARCH AND DEVELOPMENT International Business Machines Corp. Armonk, NY 10504	IBM J. Res. Dev.	INTERNATIONAL JOURNAL OF SOLIDS AND STRUCTURES Pergamon Press, Inc. Maxwell House, Fairview Park Elmsford, NY 10523	Intl. J. Solids Struc.
INDUSTRIAL RESEARCH Dun-Donnelley Publishing Corp 222 S. Riverside Plaza Chicago, IL 60606	Indus. Res.	ISRAEL JOURNAL OF TECHNOLOGY Weizmann Science Press of Israel Box 801, Jerusalem, Israel	Israel J. Tech.
INGENIEUR ARCHIV Springer-Verlag New York Inc. 175 Fifth Ave. New York, NY 10010	Ing. Arch.	JAPAN SHIPBUILDING AND MARINE ENGINEERING Technical Information Service, Inc. 2-8 Kanda-Kagi-cho, Chiyoda-ku Tokyo, Japan	J. Shipbdg. Mar. Engr.
INSTITUTION OF MARINE ENGINEERS, TRANSACTIONS Marine Media Management Ltd. Memorial Bldg., 76 Mark Lane London EC3R 7JN, UK	Instn. Mar. Engrs., Trans.	JOURNAL DE MÉCANIQUE Gauthier-Villars 55 Quai des Grands Augustines, Paris 6, France	J. de Mécanique
INSTITUTION OF MECHANICAL ENGINEERS, (LONDON), PROCEEDINGS Institution of Mechanical Engineers 1 Birdcage Walk, Westminster, London SW1, UK	Instn. Mech. Engr. Proc.	JOURNAL OF THE ACOUSTICAL SOCIETY OF AMERICA American Institute of Physics 335 E. 45th St. New York, NY 10010	J. Acoust. Soc. Amer.
INSTRUMENT SOCIETY OF AMERICA, TRANSACTION Instrument Society of America 400 Stanwix St. Pittsburgh, PA 15222	ISA Trans.	JOURNAL OF AIRCRAFT American Institute of Aeronautics and Astronautics, 1290 Ave. Americas, New York, NY 10019	J. Aircraft
INTERNATIONAL JOURNAL OF CONTROL Taylor and Francis Ltd. 10-14 Macklin St. London WC2B 5NF, UK	Intl. J. Control	JOURNAL OF THE AMERICAN CONCRETE INSTITUTE American Concrete Institute P. O. Box 4754, Redford Station Detroit, MI 48219	J. Amer. Concrete Inst.
INTERNATIONAL JOURNAL OF EARTHQUAKE ENGINEERING AND STRUCTURAL DYNAMICS John Wiley and Sons Ltd. 650 Third Ave. New York, NY 10016	Intl. J. Earthquake Engr. Struc. Dynam.	JOURNAL OF THE AMERICAN HELICOPTER SOCIETY American Helicopter Society, Inc. 30 E. 42nd St. New York, NY 10017	J. Amer. Helicopter Soc.

PUBLICATION AND ADDRESS	ABBREVIATION	PUBLICATION AND ADDRESS	ABBREVIATION
JOURNAL OF BALLISTICS 1339 Brandywine St. Philadelphia, PA 19123	J. Ballistics	JOURNAL OF SPACECRAFT AND ROCKETS American Institute of Aeronautics and Astronautics, 1290 Ave. Americas New York, NY 10019	J. Space- craft and Rockets
JOURNAL OF COMPOSITE MATERIALS Technomic Publishing Co., Inc. 265 Post Road West Westport, CT 06880	J. Composite Matl.	JOURNAL OF TESTING AND EVALUATION American Society for Testing & Materials 1916 Race St. Philadelphia, PA 19103	J. Test Eval.
JOURNAL OF ENGINEERING MATHEMATICS Academic Press 198 Ash Street Reading, MA 01867	J. Engr. Math.	LUBRICATION ENGINEERING American Society of Lubrication Engineers, 838 Busse Highway Park Ridge, IL 60068	Lubric. Engr.
JOURNAL OF ENVIRONMENTAL SCIENCES Institute of Environmental Sciences 940 E. Northwest Highway Mt. Prospect, IL 60056	J. Environ. Sci.	MACHINE DESIGN Penton Publishing Co. Penton Bldg., Cleveland, OH 44113	Mach. Des.
JOURNAL OF THE FRANKLIN INSTITUTE Pergamon Press, Inc. Maxwell House, Fairview Park Elmsford, NY 10523	J. Franklin Inst.	MASCHINENBAUTECHNIK VEB Verlag Technik Oranienburger Str. 13/14 102 Berlin, E. Germany	Maschinen- bautechnik
JOURNAL OF THE INSTITUTE OF ENGINEERS, AUSTRALIA Science House 157 Gloucester Sidney, Australia 2000	J. Inst. Engr. Australia	MÉCANIQUE APPLIQUÉE Editions de l'Academie De La République Socialiste de Roumanie 3 Bis Str., Gutenberg Bucarest, Romania	Mécanique Appliquée
JOURNAL OF MECHANICAL ENGINEERING SCIENCE Institution of Mechanical Engineers 1 Birdcage Walk, Westminster London SW1 H9, UK	J. Mech. Engr. Sci.	MECCANICA Pergamon Press, Inc. Maxwell House, Fairview Park Elmsford, NY 10523	Meccanica
JOURNAL OF MECHANICAL LABORATORY OF JAPAN (English Edition) The Government Mechanical Lab., Agency of Industrial Science and Technology, 4-12 Igusa Suginami-ku Tokyo, Japan	J. Mech. Lab. Japan	MECHANICAL ENGINEERING American Society of Mechanical Engineers 345 E. 47th St. New York, NY 10017	Mech. Engr.
JOURNAL OF THE MECHANICS AND PHYSICS OF SOLIDS Pergamon Press, Inc. Maxwell House, Fairview Park Elmsford, NY 10523	J. Mech. Phys. Solids	MECHANICS RESEARCH AND COMMUNICATIONS Pergamon Press, Inc. Maxwell House, Fairview Park Elmsford, NY 10523	Mech. Res. and Comm.
JOURNAL OF PHYSICS E. (SCIENTIFIC INSTRUMENTS) American Institute of Physics 335 E. 45th St. New York, NY 10017	J. Phys. E. (Sci. Instr.)	MECHANISM AND MACHINE THEORY Pergamon Press, Inc. Maxwell House, Fairview Park Elmsford, NY 10523	Mech. and Mach. Theory
JOURNAL OF SHIP RESEARCH Society of Naval Architects and Marine Engineers 20th and Northampton Sts. Easton, PA 18042	J. Ship Res.	MEMOIRES OF THE FACULTY OF ENGINEERING, KYOTO UNIVERSITY Kyoto University Kyoto, Japan	Mem. Fac. Engr., Kyoto Univ.
JOURNAL OF THE SOCIETY OF ENVIRONMENTAL ENGINEERS The Moding Press Ltd. 6 Conduit St. London W1R 9TG, UK	J. Soc. Environ. Engr.	MEMOIRES OF THE FACULTY OF ENGINEERING, NAGOYA UNIVERSITY Library, Nagoya University The Faculty of Engineering Furo-Cho, Chikusa-ku Nagoya, Japan	Mem. Fac. Engr., Nagoya Univ.
JOURNAL OF SOUND AND VIBRATION Academic Press 111 Fifth Ave., New York, NY 10019	J. Sound Vib.	MTZ MOTORTECHNISCHE ZEITSCHRIFT Frankh'sche Verlagshandlung 7 Stuttgart 1, Pfizerstrasse 5-7 W. Germany	MTZ Motor- tech. Z.

PUBLICATION AND ADDRESS	ABBREVIATION	PUBLICATION AND ADDRESS	ABBREVIATION
NAVAL ENGINEERS JOURNAL American Society of Naval Engineers Inc. Suite 507 Continental Bldg. 1012 14th St., N.W. Washington, D.C. 20005	Naval Engr. J.	SAE PREPRINTS Society of Automotive Engineers Two Pennsylvania Plaza New York, NY 10001	SAE Prepr.
NOISE CONTROL, VIBRATION AND INSULATION Trade and Technical Press Ltd. Crown House, Morden Surrey SM4 5EW, UK	Noise Control, Vib. and Insul.	SHIPBUILDING AND MARINE ENGINEERING INTERNATIONAL Whitehall Technical Press, Ltd. Earl House, 27 Earl St., Maidstone Kent ME 1PE, UK	Shipbldg. Mar. Engr. Intl.
NOISE CONTROL ENGINEERING P.O. Box 3206 - Arlington Branch Poughkeepsie, NY 12603	Noise Control Engr.	SIAM JOURNAL ON APPLIED MATHEMATICS Society for Industrial and Applied Mathematics, 33 S. 17th St. Philadelphia, PA 19103	SIAM J. Appl. Math.
NUCLEAR ENGINEERING AND DESIGN North Holland Publishing Co. P.O. Box 3489 Amsterdam, The Netherlands	Nucl. Engr. Des.	SIAM JOURNAL ON NUMERICAL ANALYSIS Society for Industrial and Applied Mathematics, 33 S. 17th St. Philadelphia, PA 19103	SIAM J. Numer. Anal.
OIL AND GAS JOURNAL The Petroleum Publishing Co. 211 S. Cheyenne Tulsa, OK 74101	Oil and Gas J.	SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS, NEW YORK, TRANSACTIONS Society of Naval Architects and Engineers, 20th and Northhampton St. Easton, PA 18042	Soc. Naval Arch. Mar. Engr., Trans.
OSAKA UNIVERSITY, TECHNICAL REPORTS Faculty of Technology Osaka University Miyakojima, Osaka, Japan	Osaka Univ., Tech. Rept.	S/V, SOUND AND VIBRATION Acoustic Publications, Inc. 27101 E. Oviat Rd. Bay Village, OH 44140	S/V, Sound Vib.
PACKAGE ENGINEERING 5 S. Wabash Ave. Chicago, IL 60603	Package Engr.	TRANSACTIONS OF THE AMERICAN SOCIETY OF LUBRICATION ENGINEERS Academic Press 111 Fifth Ave., New York, NY 10017	Trans. Amer. Soc. Lubric. Engr.
POWER TRANSMISSION DESIGN Industrial Publishing Co. Division of Pittway Corp. 812 Huron Rd., Cleveland, OH 44113	Power Transm. Des.	TRANSACTIONS OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS United Engineering Center, 345 E. 47th St. New York, NY 10017	Trans. Amer. Soc. Lubric. Engr.
PROCEEDINGS OF THE AMERICAN SOCIETY OF CIVIL ENGINEERS Publications Office, ASCE United Engineering Center, 345 E. 47th St. New York, NY 10017		JOURNAL OF APPLIED MECHANICS	J. Appl. Mech., Trans. ASME
JOURNAL OF THE ENGINEERING MECHANICS DIVISION	ASCE J. Engr. Mech. Div.	JOURNAL OF DYNAMIC SYSTEMS, MEASUREMENT AND CONTROL	J. Dyn. Syst., Meas. and Control, Trans. ASME
JOURNAL OF THE GEOTECHNICAL ENGINEERING DIVISION	ASCE J. Geotech. Engr. Div.	JOURNAL OF ENGINEERING FOR INDUSTRY	J. Engr. Indus., Trans. ASME
JOURNAL OF THE STRUCTURAL DIVISION	ASCE J. Struc. Div.	JOURNAL OF ENGINEERING FOR POWER	J. Engr. Power, Trans. ASME
POWER Power P. O. Box 521 Hightstown, NJ 08520	Power	JOURNAL OF LUBRICATION TECHNOLOGY	J. Lubric. Tech., Trans. ASME
PRODUCT ENGINEERING (NEW YORK) McGraw-Hill Book Co. P. O. Box 1622, New York, NY	Product Engr. (N.Y.)	TRANSACTIONS OF THE INSTRUMENT SOCIETY OF AMERICA Instrument Society of America 400 Standix St. Pittsburgh, PA 15222	Trans. Instr. Soc. Amer.
ROYAL INSTITUTION OF NAVAL ARCHITECTS, TRANSACTIONS Royal Institution of Naval Architects 10 Upper Belgrave St. London SW1X 8BQ, UK	Roy. Instn. Naval Arch., Trans.		

PUBLICATION AND ADDRESS	ABBREVIATION	PUBLICATION AND ADDRESS	ABBREVIATION
TRANSACTIONS OF THE NORTH EAST COAST INSTITUTION OF ENGINEERS AND SHIPBUILDERS North East Coast Institution of Engineers Bolbec Hall, Newcastle Upon Tyne 1 UK	Trans. North East Coast Inst. Engr. Shipbldg.	WEAR Elsevier Sequoia S.A. P. O. Box 851 1001 Lausanne 1, Switzerland	Wear
VDI ZEITSCHRIFT Verein Deutscher Ingenieur GmbH Postfach 1139, Graf-Recke Str. 84 4 Dusseldorf 1, Germany	VDI Z.	ZEITSCHRIFT FÜR ANGEWANDTE MATHEMATIK UND MECHANIK Akademie Verlag GmbH Liepssiger Str. 3-4 108 Berlin, Germany	Z. angew. Math. Mech.
VEHICLE SYSTEMS DYNAMICS Swets and Zeitlinger N.V. 347 B Herreweg Lisse, The Netherlands	Vehicle Syst. Dyn.	ZEITSCHRIFT FÜR FLUGWISSENSCHAFTEN DFVLR D-3300 Braunschweig Flughafen, Postfach 3267, W. Germany	Z. Flugwiss.
VIBROTECHNIKA Kauno Polytechnikos Institut Kaunas, Lithuania	Vibro- technika		

ANNUAL PROCEEDINGS SCANNED

INTERNATIONAL CONGRESS ON ACOUSTICS, ANNUAL PROCEEDINGS	Intl. Cong. Acoust., Proc.	THE SHOCK AND VIBRATION BULLETIN, UNITED STATES NAVAL RESEARCH LABORATORIES, ANNUAL PROCEEDINGS Shock and Vibration Information Ctr. Naval Research Lab., Code 8404 Washington, D.C. 20375	Shock Vib. Bull., U.S. Naval Res. Lab., Proc.
INSTITUTE OF ENVIRONMENTAL SCIENCES, ANNUAL PROCEEDINGS Institute of Environmental Sciences 940 E. Northwest Highway Mt. Prospect, IL 60056	Inst. Environ. Sci., Proc.	UNITED STATES CONGRESS ON APPLIED MECHANICS, ANNUAL PROCEEDINGS	U.S. Cong. Appl Mech., Proc.
MIDWESTERN CONFERENCE ON SOLID MECHANICS, ANNUAL PROCEEDINGS	Midw. Conf. Solid Mech., Proc.	WORLD CONGRESS ON APPLIED MECHANICS, ANNUAL PROCEEDINGS	World Cong. Appl. Mech., Proc.

CALENDAR

MARCH 1978

25-27 **Applied Mechanics Western and J.S.M.E. Conference**, Honolulu, Hawaii (ASME Hq.)

APRIL 1978

3-5 **Structures, Structural Dynamics and Materials Conference**, [ASME] Bethesda, MD (ASME Hq.)

9-13 **Gas Turbine Conference & Products Show**, [ASME] London (ASME Hq.)

17-20 **Design Engineering Conference & Show** [ASME] Chicago, IL (R.C. Rosaler, Rice Assoc., 400 Madison Ave., N.Y., NY 10017)

17-20 **24th Annual Technical Meeting and Equipment Exposition** [IES] Fort Worth, TX (IES Hq.)

24-28 **Spring Convention** [ASCE] Pittsburgh, PA (ASCE Hq.)

MAY 1978

4-5 **IX Southeastern Conference on Theoretical and Applied Mechanics** [SECTAM] Nashville, TN (Dr. R.J. Bell, SECTAM, Dept. of Engrg. Sci. & Mech., Virginia Polytechnic Inst. & State Univ., Blacksburg, VA 24061)

8-10 **Inter-NOISE 78**, San Francisco, CA (INCE, W.W. Lang)

8-11 **Offshore Technology Conference**, Houston, TX (SPE, Mrs. K. Lee, Mtgs. Section, 6200 N. Central Expressway, Dallas, TX 75206)

14-19 **Society for Experimental Stress Analysis**, Wichita, KS (SESA, B.E. Rossi)

16-19 **Acoustical Society of America, Spring Meeting**, [ASA] Miami Beach, FL (ASA Hq.)

JUNE 1978

30 **Eighth U.S. Congress of Applied Mechanics**, [ASME] Los Angeles, CA (ASME)

SEPTEMBER 1978

24-27 **Design Engineering Technical Conference**, [ASME] Minneapolis, MN (ASME Hq.)

OCTOBER 1978

49th Shock and Vibration Symposium, Washington D.C. (H.C. Pusey, Director, The Shock and Vibration Info. Ctr., Code 8404, Naval Res. Lab., Washington, D.C. 20375 Tel. (202) 767-3306)

1-4 **Design Engineering Technical Conference**, [ASME] Minneapolis, MN (ASME Hq.)

8-11 **Diesel and Gas Engine Power Conference and Exhibit**, [ASME] Houston, TX (ASME Hq.)

8-11 **Petroleum Mechanical Engineering Conference**, [ASME] Houston, TX (ASME Hq.)

17-19 **Joint Lubrication Conference**, [ASME] Minneapolis, MN (ASME Hq.)

26-Dec 1 **Acoustical Society of America, Fall Meeting**, [ASA] Honolulu, Hawaii (ASA Hq.)

DECEMBER 1978

10-15 **Winter Annual Meeting**, [ASME] San Francisco, CA (ASME Hq.)

CALENDAR ACRONYM DEFINITIONS AND ADDRESSES OF SOCIETY HEADQUARTERS

AFIPS:	American Federation of Information Processing Societies 210 Summit Ave., Montvale, NJ 07645	ICF:	International Congress on Fracture Tohoku Univ. Sendai, Japan
AGMA:	American Gear Manufacturers Association 1330 Mass. Ave., N.W. Washington, D.C.	IEEE:	Institute of Electrical and Electronics Engineers 345 E. 47th St. New York, NY 10017
AHS:	American Helicopter Society 1325 18 St. N.W. Washington, D.C. 20036	IES:	Institute of Environmental Sciences 940 E. Northwest Highway Mt. Prospect, IL 60056
AIAA:	American Institute of Aeronautics and Astronautics, 1290 Sixth Ave. New York, NY 10019	IFToMM:	International Federation for Theory of Machines and Mechanisms, US Council for TMM, c/o Univ. Mass., Dept. ME Amherst, MA 01002
AIChE:	American Institute of Chemical Engineers 345 E. 47th St. New York, NY 10017	INCE:	Institute of Noise Control Engineering P.O. Box 3206, Arlington Branch Poughkeepsie, NY 12603
AREA:	American Railway Engineering Association 59 E. Van Buren St. Chicago, IL 60605	ISA:	Instrument Society of America 400 Stanwix St. Pittsburgh, PA 15222
AHS:	American Helicopter Society 30 E. 42nd St. New York, NY 10017	ONR:	Office of Naval Research Code 40084, Dept. Navy Arlington, VA 22217
ARPA:	Advanced Research Projects Agency	SAE:	Society of Automotive Engineers 400 Commonwealth Drive Warrendale, PA 15096
ASA:	Acoustical Society of America 335 E. 45th St. New York, NY 10017	SEE:	Society of Environmental Engineers 6 Conduit St. London W1R 9TG, UK
ASCE:	American Society of Civil Engineers 345 E. 45th St. New York, NY 10017	SESA:	Society for Experimental Stress Analysis 21 Bridge Sq. Westport, CT 06880
ASME:	American Society of Mechanical Engineers 345 E. 47th St. New York, NY 10017	SNAME:	Society of Naval Architects and Marine Engineers, 74 Trinity Pl. New York, NY 10006
ASNT:	American Society for Nondestructive Testing 914 Chicago Ave. Evanston, IL 60202	SPE:	Society of Petroleum Engineers 6200 N. Central Expressway Dallas, TX 75206
ASQC:	American Society for Quality Control 161 W. Wisconsin Ave. Milwaukee, WI 53203	SVIC:	Shock and Vibration Information Center Naval Research Lab., Code 8404 Washington, D.C. 20375
ASTM:	American Society for Testing and Materials 1916 Race St. Philadelphia, PA 19103	URSI-USNC:	International Union of Radio Science - US National Committee c/o MIT Lincoln Lab., Lexington, MA 02173
CCCAM:	Chairman, c/o Dept. ME, Univ. Toronto, Toronto 5, Ontario, Canada		